Issues for low-emission, fuel-flexible power systems

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Abstract

Modern power generation systems can produce clean, economical energy. Gas turbines, modern reciprocating engines and fuel cells may all play a role in new power production, both for electric power and mechanical drive applications. Compared to their counterparts of even a decade ago, new power systems have significantly reduced pollutant emissions. However, the careful balance between low emissions and operating performance often requires that system performance be optimized on a single fuel. Thus, for example, a gas turbine designed to produce low emissions on natural gas may not easily achieve the same emission goals on a different gaseous fuel. This paper reviews the various issues associated with changes in gaseous fuel composition for low-emission turbines, reciprocating engines and fuel cells. © 2001 Published by Elsevier Science Ltd.

Keywords: Fuel composition; Gas turbines; Reciprocating engines; Fuel cells; Natural gas

Contents

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1. Introduction

Global interest in clean power generation has driven continued improvement of power systems. Improvement efforts focus on reducing emissions, improving efficiency and lowering costs without sacrificing reliability. Advances are occurring on all three fronts, but progress is usually achieved on a single-composition fuel. Thus, there is a need to consider fuel composition and fuel property variables on advanced power generation systems. Development programs for stationary gas turbines have highlighted this need. Operation of low-emission, high-efficiency gas turbines has been complicated by the variability in pipeline natural gas (NG). Emerging interest in gaseous fuels generated by biomass gasification or landfill sources raises yet another set of issues, because these fuels can have widely differing properties depending on changes in feedstock or specific site conditions. Development of "green" technology to utilize these fuels therefore requires a clear understanding of the role that the fuel type will play.

The possible gaseous fuel sources for stationary power cover a wide range of composition and heating values. Table 1 is a listing of fuel gases from "nontraditional" sources. It was presented by Meier et al. [1] in a discussion of gas turbines burning landfill gas. The table is included here to emphasize that achieving fuel flexibility with low emissions may include fuels with a wide range of heating value and composition. It is important to note that the heating value alone does not determine the properties of the fuel. For example, in Table 1 the refinery waste gases have roughly one-half the heating value of NG, but may have a higher flame temperature due to the hydrogen content (see Section 1.1). These differences demonstrate the complication of using current low-emission turbine combustors on these fuels. Significant modifications may be needed to achieve ultra-low emission levels. Even pipeline NG may include enough variability to compromise emission levels in gas turbines and may likewise modify engine performance on spark-ignition engines.

In this paper the role of fuel composition on gas turbines, reciprocating engines and fuel cells is discussed. Most of the discussion centers on gas turbines, primarily because low-emission turbines are enjoying significant market demand for new power. However, with expected changes in the electric utility industry, distributed power may be expected to increase demand for very small generators (less than 1 MW). This demand may be filled by reciprocating engines, fuel cells or even "micro-turbines". Due to potential increases in future demand for low-emission power systems, this paper was written to highlight the technical

Table **1**
Fuel gases from various sources from Meier et al. [1]

Source of fuel gas	Typical composition	LHV, MJ/nm^3 (Btu/scf)	Comments
Biomass: solid wastes in sanitary landfills, liquid sewage, residues from fruit and vegetable canneries, animal and crop wastes, marine plants including macro algae, water hyacinth and sea kelp	**30–50%** CH_4, CO_2, N_2, O_2	**11.81–21.66 (300–550)**	Produced by biodegradation of organic matter
GOB gas from coal mines	**30–50%** CH_4, N_2, O_2	**11.81–17.72 (300–450)**	Methane trapped in coal seams mixed with air
Raw natural gas used in gathering and reinjection	20–100% CH_4, C_2H_6, C_3H_8, C_4H_{10}, C_5H_{12}, C_6H_{14}, CO_2, N_2, He	**23.63–53.16 (600–1350)**	LHV may vary by reservoir and separator operation characteristics
Natural gas used in transportation and distribution	70–95% CH_4, C_2H_6, C_3H_8, C_4H_{10}, C_5H_{12}, C_6H_{14}, CO_2, N_2, He	**33.47–41.35 (850–1050)**	LHV variations within the same source are kept below **+10%**
Process natural gas in nitrogen rejection plants	CH_4/N_2 mixture	**11.81–23.63 (300–600)**	A waste stream of medium calorific value depending on plant design, operation, and rejection efficiency
Refinery waste gas	40–90% H_2, CH_4, C_2H_6, C_3H_8, C_4H_{10}, C_5H_{12}	**15.75–37.41 (400–950)**	Waste gas, from the platforming process, used to increase the H/C ratio in the refining of liquid fuels
Residual oil gasification	**50–55%** CO, **30–40%** H_2, CO_2, N_2, CH_4, H_2O	**9.85–13.78 (250–350)**	Gasification of residual oil into a medium-Btu gas of acceptable quality developed by Texaco
Unconventional natural gas	Similar to conventional NG	Comparable to conventional NG	Untapped natural gas resources to be recovered from tight sand formation, Devonian shale, aquifers, hydrate deposits, and ultra-deep reservoirs
Abiogenic gas	–	–	Natural gas of non-biological origin trapped at great depth in northern latitudes where little or no vegetation existed

issues associated with low-emission power generation using fuels with different properties. For readers who are unfamiliar with combustion, Sections **1.1** and 1.2 are provided as a tutorial on concepts and nomenclature that are used throughout the paper. The technical reasons for fuel sensitivity in power generation are outlined in Sections 2, 3 and **4**, for gas turbines, reciprocating engines, and fuel cells, respectively. Section **5** presents information on natural gas properties, and conclusions are presented in Section 6.

1.1. Combustion principles and terminology

Perhaps the most significant parameter used to characterize combustion is the fuel/air ratio (*f/a*), expressed either on a volume or mass basis. With precisely enough air to (theoretically) consume all of the fuel, combustion is said to have a stoichiometric *f/a* ratio. Adding more air produces combustion that is fuel-lean, and adding less air produces fuel-rich combustion. Because differing fuels have different stoichiometric *f/a* ratios, it is convenient to normalize the fuel/air ratio by the stoichiometric value, producing the

well-known equivalence ratio ϕ:

$$\phi = \frac{(f/a)}{(f/a)_{\text{stoich}}}$$

By referring to the equivalence ratio, combustion using different types of fuel is readily described as lean if $\phi < 1$ or rich if $\phi > 1$.

Another important combustion parameter is the flame temperature. Flame temperatures are determined by a balance of energy between reactants and products. In principle, the highest temperatures would be produced at $\phi = 1$, because all the fuel and oxygen would be consumed. In practice, the effects of species dissociation and heat capacity shift the peak temperature to slightly above stoichiometric ($\phi \sim 1.05$). Accurate calculation of the flame temperature for different fuels at arbitrary operating conditions can be carried out using chemical equilibrium software such the NASA CEC code [2] or similar commercial software. For many fuels of interest, correlations by Chang and Rhee [3] and Gulder [4] can be used to estimate the effect of fuel

changes on flame temperatures in a constant pressure (isobaric) environment.[3] The fuel type is important in determining the flame temperature. To provide a sense of magnitude, the list below compares calculated adiabatic flame temperatures of two hydrocarbons, carbon monoxide and hydrogen. This list applies to stoichiometric combustion in ambient air, and was taken from Lewis and von Elbe [5, Appendix D, Table 2]:

Methane	CH_4	2223 K
Propane	C_3H_8	2261 K
Carbon monoxide	CO	2381 K
Hydrogen	H_2	2370 K

Note in particular that the methane temperature is 150 K lower than hydrogen and carbon monoxide. For reasons explained later, this distinction makes it somewhat easier to produce low-emissions from natural gas (which is mostly methane) compared to syngases containing undiluted H_2 and CO.

1.1.1. Reacting flow classification: premixed flames

Reacting flows are classified in several categories: premixed flames, diffusion flames and reactor models. As the name implies, premixed flames have fuel and oxidizer mixed before arriving at the flame, and the rate of combustion can be characterized by the flame speed—essentially how fast a flame will propagate through the unburned mixture. A Bunsen burner, or gas torch are common examples of premixed combustion. The familiar flame cones observed on these burners are established by the flame speed balancing the premixed gas velocity. The balance between the flow rate and flame propagation is modulated by the flame surface area. Within certain limits, the flame surface can shrink or grow to accommodate changes in the flow rate, or changes in the flame speed due to differences in fuel type, fuel concentration or ambient temperature. If the flow rate is too high relative to the flame speed, the flame will be blown-off. At the other extreme, if the flame speed is large compared to the local gas velocity, the flame will propagate into the approach flow (i.e. flashback). Thus, the flame speed determines many of the significant features of how combustion occurs. A useful compilation of flame speed data for a variety of fuels has recently been reported by Kretschmer and Odgers [6]. Most of the hydrocarbon fuels have similar (maximum) laminar flame speeds, around 40 cm/s at standard conditions. However, hydrogen has a much greater flame speed, more than 240 cm/s at standard conditions. As discussed later, this introduces design

challenges for fuels with significant hydrogen content. Although the laminar flame speed is an important combustion parameter, most engine applications rely on turbulent flame propagation into mixtures, which are preheated by compression. The actual flame propagation speed rises significantly with premixed gas temperature and turbulence levels. These effects on flame speed can be approximately calculated from correlations described by Lefebvre [7]. Equivalence ratio also plays a significant role in the flame speed. For hydrocarbons, the flame speed is highest just above $\phi = 1$. Hydrogen and carbon monoxide are different in this regard, and have their peak flame speed near $\phi = 2$. For sufficiently lean conditions, the flame speed drops to zero, and the flame is extinguished. This is known as the weak limit. Odgers et al. [8] developed a correlation of the weak limit for a variety of fuel types, covering most hydrocarbons, hydrogen and carbon monoxide.

Another important consideration in premixed combustion is autoignition. At sufficiently high ambient temperatures, flame propagation is not needed to initiate combustion reaction. Instead, the reactions can occur spontaneously throughout the premixed gases. The time needed for initiating spontaneous combustion is known as the autoignition time. In reciprocating engines, and in premixed gas turbine combustors, the autoignition time is recognized as a critical design parameter. For reasons discussed later, autoignition is preferably avoided in both these applications because the engine is designed to produce stable combustion via flame propagation (or diffusion flames). The autoignition time depends critically on the ambient pressure and on fuel composition. As discussed in Section 2.3.2, longer hydrocarbons (greater carbon/hydrogen ratios) or fuels with significant hydrogen content tend to have shorter autoignition times, making these fuels difficult to use without uncontrolled ignition in high-pressure reciprocating engines or premixed gas turbines.

1.1.2. Reacting flow classification: diffusion flames

Diffusion flames supply both fuel and oxidizer to a reaction zone in an unmixed state. Heat is released as the two species diffusively mix and react. A reaction zone is established where the diffusion of reactants balances the generation of products and heat, thereby allowing reactions to occur with ideal stoichiometry. Thus, in this reaction zone, flame temperatures can approach the stoichiometric value, independent of excess air mixed in downstream of the reaction zone. If the reaction rate is very rapid, combustion is essentially controlled by species diffusion, and the impact of fuel type can be related to the physical properties of the fuel. For example, in industrial burners, changing fuel types can affect the length of the flame reaction zone, with hydrogen and CO flames being the shortest at comparable conditions simple because these fuels require less oxidant to complete combustion, see Turns [9]. In engine applications, the complex flow-field prevents a simple description of how the fuel type changes combustion behavior. For example, in

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[3] In reciprocating engines, the pressure rises during combustion because heat release is confined to the chamber volume, approaching constant volume (isochoric) conditions. Thus, the flame temperature is higher compared to a constant pressure environment due to the difference in specific heat for isobaric versus isochoric heat addition.

reciprocating engines where fuel is directly injected into the cylinder, reactions may begin spontaneously in regions where mixing is occurring, creating a combination of partially premixed flames, and diffusion burning. In these cases, autoignition is again a consideration, and the comments in the preceding section about autoignition time scales are relevant.

1.1.3. Reacting flow classification: stirred and plug flow reactor models

A third category for combustion is not usually referred to as a flame, but simply as a reactor. A stirred reactor represents an idealization where combustion products are back-mixed with reactants so quickly that the reaction zone is distributed uniformly in space. In an ideal stirred reactor, no gradients in temperature or species exist; the combustion region can be characterized by a single value for temperature and all species. Because all the reaction products are back-mixed with the reactants, the chemical pathways for combustion and pollutant formation can be different than in purely premixed or diffusion flames.

In practical combustors, intense turbulent mixing of reaction zones can produce back-mixing conditions that approach the stirred reactor idealization. For this reason, studies of pollutant formation and fuel chemistry are often carried out in a stirred reactor. As an example, Section 2.3.5 presents recent studies of pollutant chemistry in a stirred reactor.

The last category of reacting flow is the plug-flow reactor, In the plug-flow reactor, chemical reactions are carried along with the flow, and there is no back-mixing of product species. This corresponds approximately to what happens to the slower chemical reactions which occur downstream of the main heat release zone in turbine combustors. Early models of combustion system have used combinations of stirred and plug-flow reactors to represent the combined effect of back-mixing and plug-flow aerodynamics, see Swithenbank et al. [10].
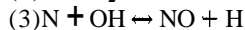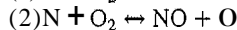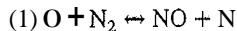
1.2. Pollutant species

There are three major pollutant species routinely considered in energy generation from combustion: unburned hydrocarbons (UHC), carbon monoxide (CO), and the oxides of nitrogen (NO and NO_2, collectively called NO_x).[4] Both UHC and CO are the products of incomplete combustion. Given sufficient time, and at high enough temperatures, these two pollutants will be further oxidized to carbon dioxide and water. In practice, completing oxidation can be a complex design issue. For example, poorly controlled cooling flows in turbine combustors can entrain

partial combustion products, quenching CO oxidation. As a second example, the piston-cylinder gap in reciprocating engines is surrounded by relatively cold metal walls, and can trap and quench UHC oxidation. Completing oxidation is simply a matter of providing enough residence time at high temperature. This becomes more difficult with fuels that produce a relatively low flame temperature. As will be seen later, attempts to use (particularly) low-heating value fuels require extended residence time to complete oxidation.
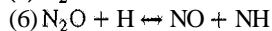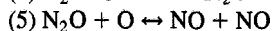
NO, pollutants are generated by oxidation of nitrogen in the combustion air, or in some instances, in the fuel. Diffusion burning occurs at temperatures high enough to oxidize atmospheric nitrogen, producing the pollutant NO, which can further oxidize to NO_2, with both pollutants collectively described as NO,. Because NO, plays a role in the production of photochemical smog and acid rain [11], stringent regulations of NO, have been established in the United States and abroad [12]. Reducing NO, emissions is a major motivation for turbine combustor development programs, including the United States Department of Energy Advanced Turbine Systems Program [13].

For later reference, the chemical mechanisms that produce NO, are listed below. These reactions represent the major pathways for NO, formation; see Nicol et al. [14] for a more detailed description of the chemical pathways. Various authors have used different names for these pathways, or include different reactions. This is a result of advances in understanding the relative importance of these mechanisms. For example, until recently, the nitrous oxide path was simply lumped as an extension of the prompt mechanism [11], but has emerged as important chemical path in lean burning gas turbines. The nitrous oxide route is now usually referred to as a distinct mechanism:

Extended Zeldovich mechanism
(1) $O + N_2 \leftrightarrow NO + N$
(2) $N + O_2 \leftrightarrow NO + O$
(3) $N + OH \leftrightarrow NO + H$
Nitrous oxide
(4) $N_2 + O + M \leftrightarrow N_2O + M$
(5) $N_2O + O \leftrightarrow NO + NO$
(6) $N_2O + H \leftrightarrow NO + NH$
Prompt
(7) $N_2 + CH \leftrightarrow HCN + N$
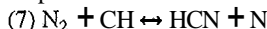
The prompt mechanism is followed by a sequence of reactions converting HCN to NO; reaction (7) is just the initiation. The detailed sequence was reported by Fenimore [15], and the prompt mechanism is sometimes referred to as "Fenimore-prompt" or just "Fenimore". The CH reaction is also important for fuels containing nitrogen which can directly form the HCN species, (see Section 2.4).

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[4] *Soot* emissions are not discussed in this paper. Gaseous **fuels** are generally free from soot emission problems.
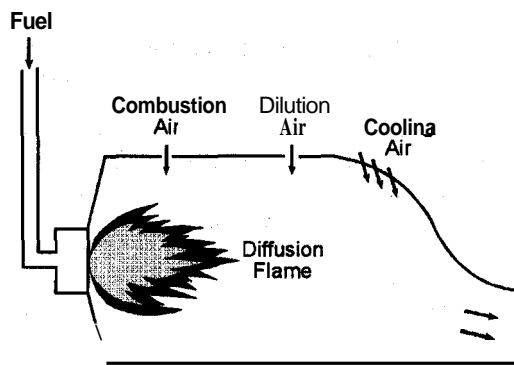
Fuel

Fig. 1. Schematic of diffusion-style gas turbine combustor. Fuel and air mix in the combustion region.

The extended Zeldovich mechanism is also known as the thermal mechanism when the O and H species are at equilibrium levels. The thermal route is a primary mechanism for NO, when flame temperatures are above approximately 1800 K (2780°F). Below this temperature, the thermal reactions are relatively slow. Thus, a common approach to NO, control is to reduce the combustion temperature so that very little thermal NO, can form.

In the absence of thermal NO, the other mechanisms become significant. Non-equilibrium concentration of O or H atoms in the flame region can produce NO, via reactions (1)–(3), and this is known as Zeldovich NO. The nitrous oxide path depends on the intermediate species N_2O which itself is generated by 0-atom attack of nitrogen. The effect of fuel composition on these various reactions is a subject of current research and will be discussed in more detail in Section 2.3.5.

2. Low-emission stationary gas turbines

As of this writing, combined-cycle gas turbines (i.e. with an attached steam cycle) burning natural gas can achieve thermal efficiencies as high as 60%. Using low-emission combustors, NO, pollutants from these engines can be reduced to less than 10 parts per million (volume basis, corrected to 15% oxygen). These very high efficiencies with corresponding low NO, emissions have required considerable advances in combustor design. Until the last decade, stationary gas turbines universally employed diffusion-style combustors, where fuel and air are injected as separate streams into the combustor primary zone (Fig. 1). The burning rate is essentially controlled by the mixing between the fuel and air streams. This approach produces a very stable flame because rapid combustion reactions are easily sustained in the high-temperature diffusion zones where mixing occurs. A second advantage for diffusion combustion is that the flame cannot propagate in the fuel

supply because no oxidant is present. Because of the flame stability, diffusion-style combustors are relatively trouble free and can readily accommodate modest changes in gaseous fuel composition.[5]

In spite of the combustion advantages, diffusion flames are poor candidates for low-emission applications. As discussed in Section 1.1.2, diffusion burning occurs at temperatures high enough to oxidize atmospheric nitrogen, producing significant quantities of thermal NO. Most of the discussion that follows will focus on efforts to reduce NO. This is not because the other pollutants (CO and UHC) are unimportant, but because the effort to reduce NO, has dominated technology development. As seen later, problems with CO or UHC emission often result from changes intended to lower NO.

2.1. Review of NO, control methods

Regulatory pressures to reduce NO, emissions have produced a number of technical approaches. Exhaust stream cleanup using selective catalytic reduction (SCR) was developed as a method to remove NO, after gases leave the turbine. The exhaust stream is treated with injected ammonia in a catalyst reactor converting NO, to molecular nitrogen. The catalytic reaction occurs in a fairly narrow temperature window, 560–670 K (550–750°F), so that SCR is usually installed in cogeneration systems where additional firing in the steam boiler is used to adjust the exhaust temperature. The peak temperature places a limit on raising steam, thereby reducing plant flexibility. In routine application, SCR is usually accomplished by injecting ammonia into the exhaust gases. A catalytic surface reaction is used to react the NO, pollutant with the ammonia to produce nitrogen. A disadvantage with SCR is that a small amount of ammonia (or other additive) may "slip" through the catalytic reactor, and can lead to ammonia emission. Although SCR is technically successful, its cost and complexity make it unattractive for some applications. Apart from the cost, SCR requires considerable space for hardware. These features are obvious drawbacks for modular distributed power systems. However, continuing refinements to SCR may add flexibility, and in some cases the SCR approach is preferred based on specific project requirements. As one example, SCR was selected as the preferred emission control at a refinery project because low NO, combustion was not possible due to the fuel gas hydrogen content [16]. A review of SCR and other post-combustion emission technology is presented by Muzio and Quartucy [17].

Aside from exhaust stream cleanup, combustion modification is the most obvious alternative for reducing NO. By

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[5] This is not true for liquid fuels as used on aeroengine applications. Because fuel volatility, viscosity and distillation range play a role in atomization, standard specifications are needed on aeroengine fuels to maintain combustor performance.
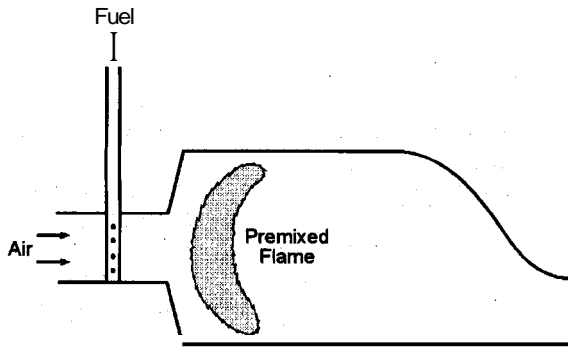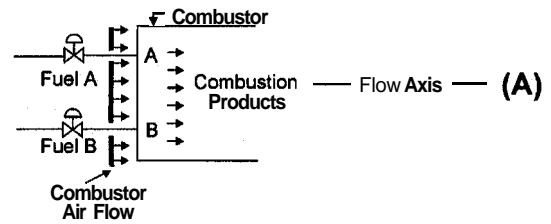
Fig. 2. Schematic of premix-style gas turbine combustor. Fuel and air mix upstream of the combustion region.

Example of parellel (or, radial) fuel staging. Fuel flow is controlled independently to injection points A and B. Note that A and B are located at the same position along the flow axis.

Example of axial fuel staging. Note that fuel is injected at two points along the flow axis. Fuel flow is controlled independently to injection points A and B.

Example of air staged combustion. Control of both the diverter and fuel valves can produce constant flame temperatures at all operating conditions.

Fig. 3. Schematic of various staging methods: (A) parallel or radial fuel staging; (B) axial fuel staging; and (C) air staging, or variable geometry.
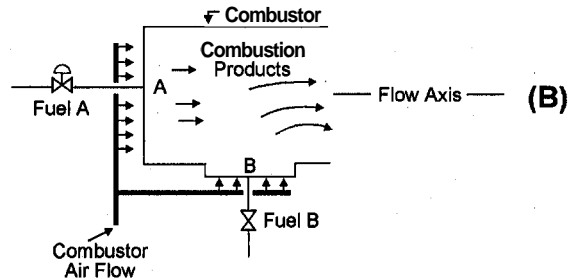
simply lowering the combustion temperature, oxidation of atmospheric nitrogen is eliminated. A direct method to lower flame temperature is to dilute the fuel. Even if the fuel participates in diffusion combustion, the diluent serves to reduce the flame temperature, mitigating NO, formation. A common diluent is water. In practice, the diluent is not premixed with the fuel, but injected co-locally with the fuel stream, so that fuel, air and diluent are mixed in the reaction zone, achieving the dilution goal without the complication of premixing the fuel and diluent upstream of the combustor. This approach is used in many utility turbine installations, and significant reductions in NO, are achievable. However, according to Correa [18], water injection will not produce the ultra-low NO, levels expected from new gas turbines. With very high levels of water injection, CO emissions and flame stability must be traded against NO, reductions. Operating experience has shown that reliability problems often accompany this method because water impurities can foul the turbine hardware. Furthermore, the approach is unattractive where clean water is scarce because the water is exhausted as steam in the stack gas.

As an alternative to water injection, most turbine vendors have developed some form of lean-premix (LPM) combustion. In LPM systems, atmospheric nitrogen (from the combustion air) plays the role of the diluent. In contrast to water injection, the term "dry low-NO," combustion is often used to describe LPM systems. As shown in Fig. 2, fuel is mixed with air upstream of the combustor at deliberately fuel-lean conditions. The fuel/air ratio typically approaches one-half the ideal stoichiometric level, meaning that approximately twice as much air is supplied as is actually needed to burn the fuel. The excess air is a key to limiting NO, formation—very lean conditions cannot produce the high-temperatures that create thermal NO,.

In principle, the LPM strategy is quite simple: keep the combustion process lean at all operating conditions. In practice, this is not so easy. If the engine is already near the limit of lean operation at full power, it is not possible to reduce the combustor temperature rise on all the fuel injectors—the flame will be extinguished. To solve this problem, some of
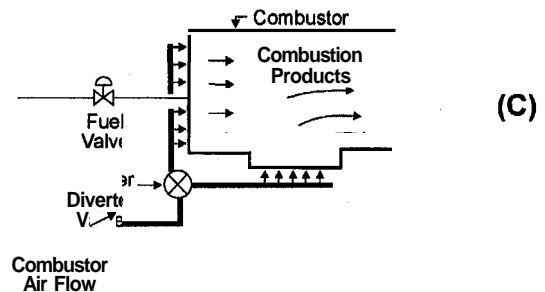
the fuel or air must be rerouted (or staged) to keep the flame within its operating boundaries. As shown schematically in Fig. 3A–C, engine developers use various methods to accomplish this goal. For a detailed discussion of these methods, see Lefebvre [19]. In brief, the fuel can be staged in two ways: radially or axially. Examples of radial (or parallel) staging include the use of pilot flames or reducing/eliminating fuel from some injectors completely (Fig. 3A). Axial staging injects fuel at two places along the combustion gas flowpath (Fig. 3B). Products from the first combustion zone are mixed with fuel and air in a subsequent combustion zone, providing an advantage for lean operation of the second zone. Finally, as shown in Fig. 3C,

some current engine designs use air staging (also known as variable geometry) to accomplish the goal of maintaining low flame temperatures [20]. This approach can maintain the desired combustion zone temperature at all operating conditions, but adds the complexity of controlling the large volume flow of combustion air.

None of the combustor configurations appear to have a proven advantage in terms of fuel flexibility. In both radial and axial fuel staging, the fuel split can be "tuned" for the various stages. Axial staging uses hot products from the first stage to feed a second stage, so that the combustion histories of gases are different in the two approaches. Parallel staging simply places a different mixture ratio in the radial direction of the combustor. A comparison of the responses of different styles of combustion to changes in fuel type has not been conducted. Premix combustion field experience, with fuel variability, is relatively recent. A recent trade journal [21] reports that premix turbine combustors have been successfully used to bum NG and refinery gas blends (variable hydrogen content, up to 40%) with less than 25 ppmv NO_x and 10 ppm CO. Although this is notable, current NO_x targets are smaller by a factor of 2.5 (i.e. less than 10 ppm) so that more work is needed to meet aggressive emission goals while incorporating fuel flexibility.

2.2. Catalytic combustion

Catalytic combustion has attracted considerable media attention for stationary engines [22]. Recent proof-of-concept tests have demonstrated very low NO_x emissions [22]. In this style of combustion, the combustion reactions are promoted by a catalytic surface at fuel-lean conditions. Again, fuel and air are premixed upstream of the combustor. Unlike other premix combustors, the fuel/air premix ratio can be deliberately established below the flammability limit so that the catalyst is necessary to initiate combustion reactions. In this manner, flashback in the premix region can be avoided, at least for the design fuel, assuming that the premixer performance achieves uniform conditions across the face of the catalyst entrance.

To date, catalytic combustors have been tested on various gaseous fuels and on vaporized liquid fuels. Vortmeyer et al. [23] studied atmospheric pressure combustion of paraffins, alcohols and vaporized liquid fuels. Excellent fuel conversion was achieved for all the fuels and with very little NO_x. The maximum combustor exit temperature was 1673 K (2551°F), which is slightly lower than current engine technology. Nevertheless, the catalyst could oxidize the range of fuels, suggesting that some fuel flexibility is possible with a catalytic combustor. Kolaczkowski [24] reviewed numerous tests of catalytic combustors and noted only that pure methane is typically the most difficult hydrocarbon to catalytically ignite. Natural gas often includes several percent ethane, so that it may be necessary to field test catalytic combustor performance for the rare applications where the NG is essentially pure methane. Kolaczkowski comments

on possible catalyst degradation from gas pipeline impurities, and notes that long-term endurance tests should include trials with varying supplies of NG.

Although catalytic combustors may oxidize a range of hydrocarbons, the fuel must be evenly premixed upstream of the catalytic reactor. This places a restriction on fuel flexibility. As explained later, the fuel Wobbe index will affect premixer effectiveness for fuel injectors that rely on jet penetration. If premixing is compromised, it is possible to locally overheat the catalyst. Schlatter et al. [25] report that premix non-uniformity of less than 10% is desired for optimal catalytic combustor performance. Thus, changes in fuel type should account for changes in premixer performance that might exceed this level. For non-recuperative engines, part-load operation requires a prebumer using approximately 10–12% of the total fuel to preheat the inlet air to the so-called light-off temperature. To avoid NO_x from this preburner, Schlatter et al. suggest using a premixed combustor upstream of the catalyst. Because this preburner uses only a small portion of the fuel, the prebumer does not need to produce ultra-low NO_x and may avoid some of the fuel sensitivity issues described below. Advanced, recuperated engines may not need a prebumer because of the higher combustor inlet air temperatures [26].

Catalytic combustion can avoid flashback problems associated with other approaches to premix combustion. If the premixer is performing correctly, the mixture approaching the catalyst may be below the flammable limit. This feature makes flashback impossible. However, at the point of fuel injection, where fuel and air are *being mixed,* the gases are indeed flammable. In this region, autoignition could occur, just as in any other style of premix combustor. Although no published reports describe this issue to date, it seems to be a prudent consideration for catalytic combustion applications where operating conditions and fuel type may promote auto-ignition. The presence of long hydrocarbons in NG supplies can promote autoignition, and this should be evaluated for potential applications.

2.3. Issues for premix combustion with fuel variability

In this section, we discuss some potential issues for premix combustion when the fuel composition is variable. Several issues must be considered flashback and flame anchoring, autoignition, static and dynamic stability, and achieving emissions goals.

2.3.1. Flame anchoring and flashback

Although lean conditions are beneficial for NO_x reduction, LPM operation introduces some complications to the combustion process. Because the fuel and air are mixed before arriving at the combustion chamber, the flame position is not constrained by mixing between the fuel and air stream. Instead, the combustor aerodynamics are designed to produce a recirculation zone where bumed products are shielded from the surrounding high-velocity gases. These

hot products serve as an ignition source for the premixed gases, and anchor the flame in the combustor. The premixed gas is burned by flame propagation from the recirculation zone into the supplied flow. The flame "surface" establishes itself at points in the flow where the local gas velocity exactly balances the flame propagation speed. In principle, describing the flame position is a simple matter of determining the turbulent flame speed and the local gas velocity. However, this process is greatly complicated by the actual conditions in the engine. Even if the turbulent flame speed were known exactly, the velocity boundary layers near the fuel injector perimeter produce a slow-speed region, where flame can propagate back into the fuel injector. Unfortunately, if the flame travels into the fuel injector, combustion may be sustained in wake regions where the fuel has not yet mixed with the air (i.e. diffusion combustion) producing very high temperatures in the premixer. The resulting thermal failure needs no further elaboration. Even without a complete flashback, the change in the fuel type can reposition the flame **so** that the fuel injector hardware exceeds design temperature specifications.

A recent paper by Kretschmer and Odgers **[6]** reports laminar flame speeds for various fuel-lean mixtures. Although the correlation of available data is helpful, it cannot be used to directly predict the flame position in a practical combustor, because the flame front is highly turbulent. As noted by Polifke et al. **[27],** practical turbine combustion can span several different turbulence regimes and models may need to include both chemical kinetics and turbulent mixing rates to calculate the heat release rate; see also Polifke et al. **[28]**. **This** greatly complicates the prediction of the reaction zone and makes it difficult to predict the precise location of the flame front. It may be useful to conduct experiments to demonstrate the parametric sensitivity of flame speed/flame position as a function of the fuel composition, but such a study would need to be conducted cognizant of the actual flame conditions in a gas turbine. Because specific application of the results will depend on the local turbulence properties in a given engine, it may be difficult to generalize the observations.

An additional complication for flame flashback arises from the potential for oscillating combustion. Changes in fuel type may lead to unexpected combustion oscillations (see Section **2.3.4).** These oscillations can be severe enough to momentarily reverse the flow in the fuel nozzle, bringing reacting flow right into the premixer. This has the potential to anchor the flame at the point of fuel injection, again damaging the fuel injector. The process was demonstrated experimentally by Keller et al. **[29]** showing that, in some cases, a step-stabilized flame could flashback via the large-scale motion produced by acoustic disturbances. The same problem was studied using numerical simulations by Najm and Ghoniem **[30]** and more recently by Thibaut and Candel **[31]**. Both of these simulations employed a simplified model of the flame, either artificially thin or thick, making it difficult to account for the effect of fuel properties on the actual flame motion.

2.3.2. Autoignition

Premixing fuel injectors must be designed to avoid autoignition. As the name implies, autoignition occurs when the premixer conditions are suitable to initiate self-sustaining combustion, even if no flames were present downstream. For example, if the premix air temperature is high enough, then simply injecting fuel into the air will begin combustion. Again, this raises the possibility of overheating and damaging the fuel injector, with subsequent damage to rotating hardware downstream.

To avoid autoignition, most premixing fuel injectors are designed around the characteristic autoignition time scale. At conditions typical of stationary engines, the autoignition time for methane (the main constituent of NG) is longer than the residence time in the premix passage, **so** autoignition can be avoided. However, it has been shown that various hydrocarbons in NG significantly lower the autoignition time, producing unexpected problems. Spadaccini and Colket **[32]** present an extensive analysis of autoignition data for fuels similar to NG. The autoignition data include pure methane and methane/hydrocarbon mixtures similar to NG. Remarkably, the autoignition time scale can change by as much as a factor of seven depending on the type of NG. The variation is mainly due to the ethane content of the different NG fuels, which can range from negligible levels to as much as 15% for certain NG supplies. Although these authors present a correlation of the available data and model predictions, the results are restricted to temperatures of more than **1200** K **(1700°F);** somewhat higher than exists in most stationary engine premixers. The correlation also lacks supporting data at lean operating conditions, which are relevant to premix combustors. The authors note that "extension of the correlations to lower temperatures is a worthwhile area for future research". Recognizing these limitations, the correlation of available data is:

$$t = 1.77 \times 10^{-14} \exp\left(\frac{18,693}{T}\right)[O_2]^{-1.05}[CH_4]^{0.66}[HC]^{-0.39}$$

$$(1)$$

where t is ignition delay time in s and T is temperature (K). Quantities in brackets are concentrations in moles per cubic cm, and the HC concentration represents all the non-methane hydrocarbons. Notice that the correlation indicates that the autoignition time is smaller at lower methane concentrations, i.e. at leaner mixtures. This counterintuitive result is attributed to the lack of data at lean operating conditions. In a homogenous mixture, the autoignition time should increase with very lean conditions, **so** that Eq. **(1)** should not be used to extrapolate to lean conditions.

Note that this correlation predicts (indirectly) a pressure dependence that is the **sum** of the concentration exponents, i.e. $P^{-0.78}$. This pressure dependence is very similar to the value reported by Odgers and Kretschmer **[33]** for vaporized aviation fuels. There is some debate about the exact value of the pressure dependence; for example, Spadaccini and TeVelde **[34]** report a pressure dependence of P^{-2} for
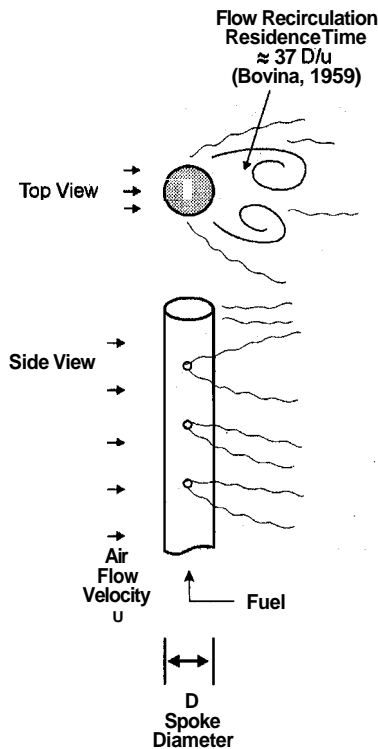
Fig. 4. How recirculation behind a fuel injection spoke.

No. 2 diesel. Whatever the exact dependence, it is well understood that autoignition time scales are reduced at elevated pressure. This pressure dependence has significant implications for fuel injectors that are designed for very high-pressure ratio engines, such as stationary aeroderivative engines. In these engines, both high compressor discharge temperatures and pressures are present. The implication is that autoignition will be a more serious problem for premix combustion in this style of engine. There are some exceptions to this generalization. For proposed humid air turbines, the humidification process will lower the combustor inlet temperature, so that it may be possible to premix without danger of autoignition.

Cowell and Lefebvre [35] measured the autoignition time for methane, propane, ethylene and acetylene. Test conditions were typical of stationary gas turbines: 670–1020 K (750–1375°F) and pressures from 1 to 10 atm. These authors reported that ignition delay was approximately an order of magnitude lower for propane than pure methane at the same operating conditions, emphasizing the critical role of fuel type on autoignition.

It is instructive to consider a specific example to understand the uncertainties associated with autoignition problems. A typical premixer may use a flow velocity of 30 m/s to avoid flashback, with length 0.15 m (6.0 in.), producing a bulk residence time of $0.15/30 = 0.005$ s (5 ms). We might conclude that autoignition could be

avoided for all fuels that have an autoignition time scale longer than 5 ms. There are two problems with this conclusion.

First, it is usually inadequate to estimate the premixer flow time based on bulk flow. Premixers usually inject fuel from spokes or holes in the wall of the premixer passage. Secondary flows behind these injection points inevitably lead to recirculation zones. As an example, Fig. 4 shows the flow around a cylinder representing a fuel injection spoke. The wake recirculation behaves as a small stirred reactor. Bovina [36] showed that the *mean* residence time for fluid particles is 37D/U, where D is the diameter of the fuel spoke and U is the free stream velocity (30 m/s in this example). Assuming a very small fuel spoke ($D = 3.2$ mm (0.13 in.)), the mean residence time in the recirculation zone is 4 ms, which is comparable to the bulk residence time. This simple example emphasizes the importance of secondary flows in the premixer. Without a description of the premixer flow time scales, it is difficult to predict whether a change in fuel autoignition time will lead to problems in a given application.

A second difficulty in predicting autoignition behavior is the lack of available data at conditions relevant to gas turbines. As already discussed, the review by Spadaccini and Colket [32] is restricted to temperatures higher than found in most gas turbines. Cowell and Lefebvre [35] covered the most relevant range of conditions, but the data are limited to 10 atm pressure and do not consider mixtures of methane and ethane (the major components of NG). Recent studies on NG autoignition in diesel applications consider these mixtures. However, the test conditions may not match those occurring in gas turbines (the temperature and pressure are higher). Naber et al. [37] compared the diesel ignition of methane, methane–ethane and so-called "peakshaving" gas (discussed in Section 5). The lowest temperatures studied were 1000 K (1340°F) at a pressure of 68 atm. At this condition, measured ignition delays ranged from 7 to 12 ms, depending on the fuel type. Lower pressures and temperatures found in gas turbine premixers should have longer autoignition times, but the physical processes noted in the next paragraph may make it difficult to use measured delay times with confidence.

Even when an accurate measurement of autoignition time is available, it is important to note that in real fuel injectors, physical processes may also play a role in determining the autoignition. Bi and Agrawal [38] consider both the fluid mechanic mixing and chemical ignition delay that occur in a diesel engine environment. Gerk and Karagoeian [39] show how fluid mechanic diffusion in shear layers can play a role in determining the autoignition time. The authors show that the process of mixing a two-dimensional fuel jet with a surrounding air stream may extend the ignition delay time. The extended time is the result quenching by the surrounding gases. In some unusual cases, the ignition may be assisted by favorable diffusion of heat and oxidizer to the interior of the fuel strip. Although the analysis is not

intended to provide quantitative values of autoignition time, the observed physics should be recognized as a contribution in the autoignition process. From a practical standpoint, the physical conditions in a premixer include diffusion mixing of the fuel and air stream. **This** is a different environment than carefully designed premix studies of autoignition time, *so* it is not possible to use published autoignition times with absolute confidence. Even where autoignition is prevented for test fuels, premixer design may need to account for fuel impurities including compressor lube oils, pipeline cleaning solvents and peakshaving gases. As noted in connection with Eq. (1), the presence of heavy hydrocarbons will shorten the autoignition time, such that variations in hydrocarbon content should be controlled. Where these complications cannot be avoided, Becker et al. [40] suggest that thermocouples should be placed in the fuel injector to monitor for upset conditions that could damage the fuel injector. Wilkes and Dean [41] propose that gas turbine fuels should be carefully monitored for dewpoint condensation of heavy hydrocarbons as a means to detect unexpected fuel impurities.

2.3.3. Static flame stability

Another important consideration for premix combustors is combustion stability. Combustion stability is divided into two classes: static and dynamic. Static stability involves where the flame is physically anchored in the combustor and how well it can be maintained during upsets or at extremely lean operating conditions. In aeroengines, the concept of flame "blowout" is well appreciated and must be avoided during conditions of rapid power changes, etc. In stationary engines, blowout has traditionally been less of a concern because earlier diffusion-style combustors were fairly forgiving in the comparatively modest operating regime of stationary engines. However, the current trend to operate stationary engine combustors very close to the lean-blowout limit has raised the status of **this** issue. A change in fuel type may not allow the engine to operate at the same lean equivalence ratio. Instead, part of the fuel may need to be "rescheduled" to a pilot flame, producing an NO_x penalty.

As another example, during rapid load shedding, the combustor must suddenly accommodate a reduction in fuel, even though the engine is operating near the lean blowout. Again, the solution is typically to use a pilot flame with the fuel control schedule adjusted to favor the pilot during upsets. For fuels with variable composition, the control schedule may require modification to accommodate the different fuel properties at different sites, because the actual blowout limit may change with the fuel properties.

To determine how changes in fuel composition will change static stability may require understanding how the flame is anchored in a specific combustor. Durbin et al. [42] showed that the choice of co-swirl versus counter-swirl fuel nozzles produced a significant difference in the lean-blowout equivalence ratio (ϕ_{lbo}). The difference was most notable for the smaller swirl vane angle (30 versus 45°), and

a pronounced difference in ϕ_{lbo} was observed between propane and methane fuels. These results were not obtained in a premix combustor, but suggest that fuel type is an important consideration for combustors operating near the lean-blowout equivalence ratio. Few papers have reported blowoff data for swirl-stabilized **LPM** combustors, but numerous papers report on stirred reactors and bluff-stabilized flames used, for example, in thrust augmentors. Lefebvre [7] showed that the blowoff velocity of bluff-stabilized flames is proportional to the square of the *laminar* flame speed. Therefore, the effect of changes in fuel properties might be anticipated by simply recognizing the associated change in laminar flame speed. In **this** regard, the paper by Kretchsmer and Odgers [6] may provide useful flame speed data for various gaseous fuels.

Hermanson et al. [43] present a fundamental study of static flame stability in a premix burner, using a core flow and lean coflow arrangement. This is similar to a radially staged fuel injector, except the flame was not swirl stabilized. These authors showed that the lean blowoff of the main (core) flow could be improved with a very lean co-flow, assuming that the core and co-flow could adequately mix. Some aspects of the blowoff limit were again successfully correlated with a model using the *laminar* flame speed. Although these authors studied only one fuel composition, knowledge of the flame speed for different fuels may again provide guidance for considering the blowoff behavior of various fuels. In a similar fashion, blowoff data from stirred reactors may be useful to understand the effect of changes in fuel type. Sturgess et al. [44] showed that a step-stabilized propane flame behaved like a stirred reactor near lean blowoff. **This** is consistent with the recognized role of the flame speed because Clarke et al. [45] showed that the stirred reactor blowoff could likewise be related to the flame speed.

2.3.4. Dynamic stability

Even where excellent static stability is achieved, dynamic stability may be a problem in low-emission engines. Pressure oscillations driven by combustion have been a repeated problem in the development of low NO_x premix combustors. Again, the emergence of **this** problem in low-emission applications is the result of specific changes made to the combustor to accommodate the premix approach. In premix systems, most of the combustion air is sent through the fuel injector, eliminating the need for downstream combustion air holes typical of diffusion-style combustors. Unfortunately, these downstream air holes provide acoustic damping that reduces the likelihood of oscillations. Furthermore, the distributed reaction-associated with diffusion burning is unlikely to synchronize heat-release perturbations with the acoustic perturbations. However, in premix systems, slight disturbances in pressure create immediate changes in the airflow, producing a subsequent change in reaction stoichiometry. Near the lean combustion limit, even minor changes in reaction stoichiometry can lead to significant variations in heat release. If these variations are synchronized with the

$$\text{Heat Input} = KA \; \underbrace{\frac{LHV}{\sqrt{S.G.}}} \; \sqrt{\frac{T_{std}}{T}} \; \sqrt{P_1 (P_1 - P_2)}$$
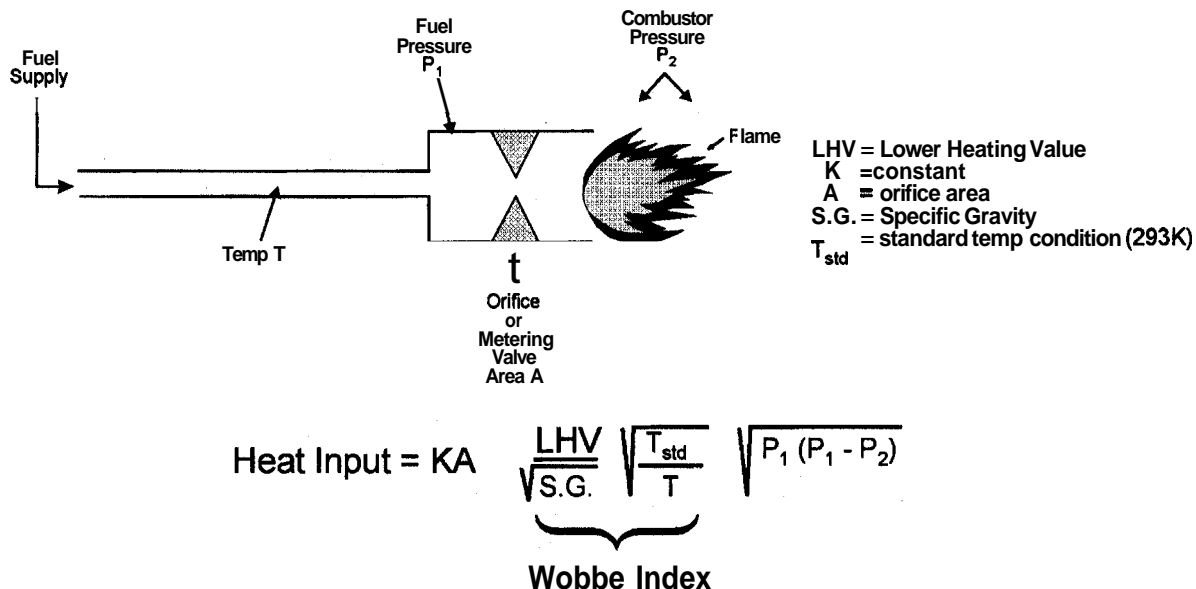
Wobbe Index

Fig. 5. Definition of the Wobbe index.

resonant pressure field, oscillating combustion can be sustained with a frequency from tens to thousands of cycles per second. Fairly exact timing is needed to synchronize the perturbations between pressure and heat release in a manner that will sustain an oscillation. This timing is affected by minor geometric changes in the fuel injector or combustor interior. In addition, it has been shown recently that the timing also depends on the specific fuel reaction time scale [46]. In a study of pulsating combustion, Kushari et al. [47] showed that the chemical reaction time scale plays a significant role in the phase between pressure perturbations and heat release, demonstrating that changes in fuel type can be expected to change dynamics. This was true for a pulse combustor with a relatively low frequency (50 Hz) and a long acoustic period (20 ms). For combustion systems with higher acoustic frequencies, say 500 Hz, the relevant acoustic period is even shorter (2 ms), meaning that simply changing the fuel type may unexpectedly align the pressure and heat release to produce (or silence) oscillations. Janus et al. [46] demonstrated this behavior in a laboratory combustor when switching between NG and propane. Practical field tests have likewise shown that the variation in pipeline NG will affect engine dynamics [48].

It is essential to eliminate pressure oscillations in commercial engines. With sufficient amplitude, the vibration and enhanced heat transfer that accompany these oscillations can lead to fatigue failure of the combustor liner in a matter of hours. To further complicate matters, the occurrence of even momentary oscillations can serve as a trigger to cause flashback, as discussed earlier. It is often possible to control oscillations by simply adding fuel to the pilot flame, although this is not a universal solution, and it again produces a NO, penalty. Several recent papers have

discussed methods to control oscillations by both passive [49] and active [50] techniques, but a universal solution to this problem has not yet been proven. At the present time, "mapping" dynamic regimes, as described by Richards and Janus [51], may provide some insurance that a given engine design is operating well inside the stable operating regime, so that minor changes in fuel type do not produce oscillations.

In closing the discussion of dynamic stability, it is noted that catalytic combustion has shown some advantages for stable combustion. For tests conducted to date, there has been no mention of problems with combustion dynamics [52]. The catalyst passages provide thermal inertia and acoustic damping to reduce the variation in heat release and acoustic energy. Both of these factors will improve dynamic stability. However, as noted above, some catalyst applications will incorporate preburners, which again must be stable.

2.3.5. Meeting emissions with fuel variability

For a low-emission application, the combustor must meet planned emission levels over the engine operating envelope. As already described, NO, control is usually achieved by simply reducing the flame temperature via fuel/air premixing. Because the engine control system adjusts the fuel/air ratio to produce a desired set-point firing temperature, a completely premixed system will have a constant flame temperature, independent of fuel type. At first glance, it would seem that the fuel composition should have an insignificant effect on NO, formation. In fact, field experience has shown that fuel properties play an appreciable role in NO, levels. There are several reasons for this. First, the fuel chemistry can affect the formation of NO, by routes other
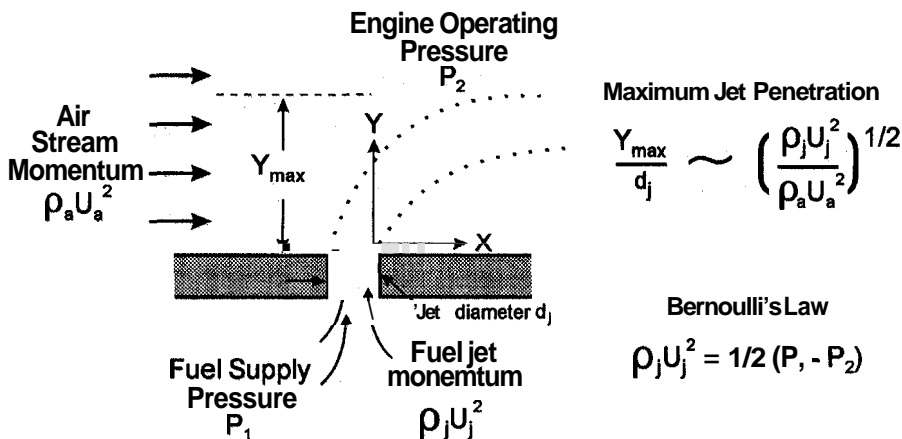
Fig. 6. Penetration of a fuel jet into an air stream.

than simple "thermal" NO. At very low NO, levels, the NO, that is produced does not originate solely from high-temperature oxidation of nitrogen but also follows the so-called N_2O route [14] and the "prompt" route [15]. Blust et al. [53], Lee et al. [54], Capehart et al. [55] and Steele et al. [56] have studied the effect of fuel composition on the emission of NO, from a stirred reactor. Their results show a small increase in NO, with fuel carbon number. These authors suggest that the change in fuel chemistry may increase intermediate species such as 0-atoms, which directly attack N_2 in the thermal route, and also the CH radical, which participates in the prompt mechanism. Imperfect premixing may couple with changes in fuel chemistry to further alter the radical pool that produces NO. In addition, mixing non-uniformities or diffusion piloting will produce some combustion regions that are hotter than the set-point firing temperature. In these regions, the deviation from the set-point firing temperature will depend on the fuel composition. Gulder [4] and Chang and Rhee [3] summarize the effects of hydrogen/carbon (WC) ratio on flame temperatures. Generally, the flame temperature drops with the WC ratio such that methane has a flame temperature which is lower than other gaseous hydrocarbons. Thus, compared to methane, the non-methane fuel components in natural gas may produce slightly higher deviations from the set-point temperature in poorly mixed regions of the combustor. This would presumably result in more NO, via the thermal route. The extent of this effect depends on the degree of mixing in the combustor, and has not yet been documented.

A second effect of fuel composition on NO, comes from the heating value per volumetric flow rate of various fuels. Premixer design is typically based on penetration of fuel jets into the nozzle air stream. If the volumetric heating value of two fuels is substantially different, then the volume flow of fuel needed to achieve a given heat input is also different. The resulting mixing jets have different velocities, producing significant change in jet penetration and mixing performance.

The change in premix performance with fuel type can be accounted for by simply maintaining the fuel Wobbe index. The concept is shown schematically in Fig. 5. If two different gaseous fuels have the same Wobbe index, they can be supplied to fixed metering runs at the same fuel pressure P_1 and still produce the same heat output at a given combustor pressure P_2. The equation for the heat input can be derived by multiplying the standard orifice flow equation [57] by the fuel lower heating value (LHV).[6] As shown, this leads to the definition of a temperature-corrected Wobbe index defined as:

$$\text{Wobbe index} = \frac{\text{LHV}}{(\text{S.G.})^{0.5}} \left(\frac{T_{\text{std}}}{T} \right)^{0.5}$$

where LHV is the lower heating value, S.G. is specific gravity of the fuel, T_{std} is standard temperature condition in Kelvin (293 K) and T is the fuel temperature in Kelvin.

Referring to Fig. 5, if two fuels have the same Wobbe index, they will produce the same heat input for corresponding pressures P_1 and P_2. In a gas turbine, P_2 is the engine operating pressure. Thus, at a given pressure P_2, two fuels with the same Wobbe index will require the same fuel pressure P_1 for engine operation. It is next shown that the same pressure P_1 will produce approximately the same fuel-mixing profiles for the two different fuels.

Most premixers operate using fuel jets penetrating into a cross-flow air stream. The mixing process between fuel jets has been studied by Hautman et al. [58] and was previously summarized by Lefebvre [7]. Referring to Fig. 6, the maximum jet penetration Y_{max} is governed by the square root of the momentum ratio between the fuel jet and the air stream. As shown, the fuel jet momentum flux can be approximated from Bernoulli's law, so that the jet momentum is directly related to the pressure drop across the fuel injector. As noted

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[6] This notation does not exactly follow Fox and McDonald [57], but for simplicity incorporates several constants in the term K shown in Fig. 5.

above, for two fuels with comparable Wobbe index, and at an engine operating pressure P_2, the same heat release can be achieved at the same fuel pressure P_1. Thus, both fuels will have the same jet momentum flux, and Y_{max} will be approximately unchanged between the two fuels. Conversely, operating a premix combustor on fuels with different Wobbe indices will likely change the premixer mixing profile and changes in emission performance may occur.

The preceding discussion of the chemical and physical features affecting NO, formation has not taken into account the interaction with other combustion requirements. As already discussed, changes in fuel type may change the lean-blowout limit, or dynamic stability of a combustor. To maintain overall lean operating conditions with a different fuel, it may be possible to increase or add pilot combustion, i.e. to "re-tune" the fuel schedule. While this may achieve the goal of maintaining stable combustion, it may also compromise the ultra-low NO, performance.

2.3.6. Medium heating-value fuel

There is no formal definition of what constitutes a medium heating-value fuel, but "medium" is a relative measure compared to commercial NG. In the literature, heating values in the range 8–30 MJ/m^3 (200–800 Btu/scf) have been classified as medium heating-value fuels. Compared to NG, it might seem to be easier to reduce NO, emissions from medium heating-value fuels because the name (mistakenly) implies that the flame temperature and thus, the thermal NO, should also be lower. Depending on the composition of the fuel, this is not generally true. A common source for medium heating-value fuels is oxygen-blown gasification of coal or residual oil. Because these gases are "manufactured" from other fuels, they are commonly referred to as synthesis gas, or syngas fuels. For these fuel the fuel gas contains significant quantities of hydrogen and carbon monoxide. Compared to NG, the stoichiometry of these gas mixtures requires a smaller volume of air for complete combustion, producing greater flame temperatures. As a further complication, hydrogen has a very high flame speed and very short ignition delay. Thus, it is very difficult to avoid flashback or autoignition in a premixed burner. The standard approach to premixing is unlikely to work for these fuels.

Dobbeling et al. [59] discuss the challenge of building a low NO, burner for these fuels. The flame temperatures can be reduced by nitrogen or water injection in the syngas, but significant dilution requires adding high-quality water or nitrogen from an air separation plant and then controlling large volumes of syngas. In an effort to reduce the need for dilution, these authors modified the well-known ABB "double-cone" burner to inject syngas from the perimeter of the nozzle exit directly into the flame recirculation zone. The intent was to inject the fuel into the vortex breakdown region of the flow where mixing may be rapid enough to prevent diffusion combustion. The results were partly successful. Modest NO, emissions (25 ppmv at 15% O_2)

could be achieved with some nitrogen dilution, but the undiluted syngas still produced fairly high emissions (120 ppmv NO, for simulated coal-gas). **An** interesting observation was that atmospheric pressure testing was a poor indicator of the best fuel nozzle geometry. Optimization tests carried out at 1 atm did not indicate the best fuel injector configuration studied at turbine pressures.

More recently, Dobbeling et al. [60] studied the premix combustion of a medium heating-value fuel with a composition of 30% hydrogen, 60% CO and 10% nitrogen. This particular fuel composition is typical of oxygen-blown coal gasification. NO, emissions less than 10 ppmv were recorded at some operating conditions and CO was always less than 100 ppmv, although the authors indicated a need to conduct additional traverse measurements to validate these reported emission levels. To avoid flashback or autoignition problems, these authors used a (bulk) fuel nozzle velocity of 75 m/s at all times, producing an average nozzle residence time of less than 4 ms. For the intended compressor discharge conditions (14 bar, 620 K), the dynamic head at this velocity is 1.6% of the compressor discharge pressure, contributing to the pressure loss as the premix gases enter the combustor. The penalty to avoid flashback with these fuels is added pressure drop accompanied by slightly lower system efficiency.

The preceding discussion has focussed on medium heating-value fuels that have a higher flame speed and flame temperature than NG, particularly due to the presence of hydrogen. In contrast, some medium heating-value fuels are much simpler to handle because of a high diluent level. For example, the primary composition of landfill gas is methane diluted with carbon dioxide. In this case, the maximum flame temperature is lower than in conventional NG, so it should be relatively easy to produce low NO, emissions. Meier et al. [1] discussed the changes needed to handle landfill gas in conventional, diffusion-style gas turbine combustors. Depending on the specific heating value of the fuel, it was possible to operate the turbine on medium heating-value fuels with some modifications to the turbine operating schedule, fuel injectors, etc. For the medium heating-value fuel (CH_4/CO_2, with a lower heating value of 17.1 MJ/m^3 (458 Btu/scf)), NO, emissions were less than 20 ppmv over the entire engine load range even though the combustor was a diffusion-style combustor. The authors did not report the corresponding CO and UHC, but cautioned that both of these emissions may be difficult to correlate with engine operation. Although this paper predates current low-emission premix combustors, many of the issues that would arise in adopting a premix combustor to burn diluted medium heating-value fuels are discussed. The authors note that the flame stability characteristics, turbine temperature pattern factor and turbine/compressor balance must be considered before switching to a fuel that requires a different volume flow rate for the same heat input. Likewise, maintaining ultra-low emissions for variable fuels requires maintaining the fuel injector

mixing profile, which will again change with the fuel Wobbe index (see Fig. 5).

2.3.7. Low heating-value fuel

Although there is no formal definition of what divides low from medium heating value, the published literature distinguishes "low" from "medium" at around 8 MJ/m^3 (−200 Btu/scf). For gas turbines, the usual source of low heating-value fuel is air-blown gasification of coal or, more recently, biomass gasification. A significant feature of low heating-value fuels is that they often contain ammonia produced during the gasification of fuel-bound nitrogen in coal or biomass. Fuel ammonia greatly complicates NO, reduction; this issue is discussed in detail in Section 2.4. Because of the high dilution level, these fuels have lower flame temperatures and lower flame speeds than NG or medium heating-value fuels with hydrogen. From the standpoint of thermal NO, emissions, this is an advantage. For example, Kelsall et al. [61] studied a combustor designed to burn gaseous fuels having a heating value of approximately 4 MJ/m^3 (~100 Btu/scf). In the absence of fuel nitrogen (ammonia), NO, emissions as low as 5 ppm were recorded with turbine inlet temperatures as high as 1645 K (2500°F). This low NO, level is directly attributed to the low flame temperature. It should be noted that the authors avoided high CO emissions by reducing the combustor loading. These authors point out a number of the difficulties in handling a very low heating-value fuel. Because the volume of the fuel flow is so great, the combustor aerodynamics are significantly affected by the fuel flow and must be designed accordingly. This may handicap low-emission backup operation on conventional fuels. Kelsall et al. [61] were able to operate the same combustor at 8 bar on NG, producing 35 ppm NO,. Although this is much greater than current low-emission targets, further optimization might reduce emissions on both fuels. The ability to operate on a standby fuel is a consideration for applications where the gasifier may need maintenance or repairs, but the turbine power must continue uninterrupted.

It is important to understand that low heating-value fuels will require combustors with adequate residence time to achieve CO oxidation. Because of the low flame temperatures, thermal NO, may be less of a consideration than CO oxidation. For example, Neilson et al. [62] report on an aeroderivative combustor modified to burn low heating-value fuel. The intended source of the fuel is air-blown biomass gasification. Note that the combustor residence time is typically very short in aeroderivative engines, so that CO oxidation may be compromised. These authors showed reasonable NO, emissions on biomass gas (less than 30 ppmv at 15% O$_2$), but relatively high CO emissions (between 100 and 600 ppm at 15% O$_2$), and they suggest that exhaust stack scrubbing or additional emissions modifications may be needed to reduce the CO levels.

Luessen [63] summarizes the issues associated with burning LHV fuels in gas turbines and reports on some of the experience gained from various coal gas and steel mill gas applications. Although apparent that these fuels can be reliably used in turbines, the paper does not comment on specific requirements needed to produce ultra-low emissions performance.

2.3.8. Dual-fuel requirements

In some turbine applications, the ability to operate on a standby fuel can provide a significant reduction in the fuel contract cost. Turbine operators who require uninterrupted gas supply must typically pay a premium to secure guaranteed service. This provides an incentive to design combustors capable of burning both gaseous and standby liquid fuels. Although it is relatively easy to add dual-fuel capability to a gas-fueled combustor, it is not so easy to produce ultra-low emissions on liquid fuels. In principle, low emissions could be achieved if the fuel were completely vaporized and mixed prior to combustion, so that the benefits of premix combustion could be realized. This approach is typically referred to as lean-premix-prevaporize (LPP). In practice, the process of atomization usually produces a small number of large fuel drops that do not evaporate prior to arriving at the main flame front. These drops burn locally near stoichiometric conditions, contributing to thermal NO,. Even where droplet evaporation is complete in the premixer, the problem of achieving uniform mixing of the fuel vapor and air stream is exacerbated by the restricted trajectories of the fuel spray (i.e. the spray should not impinge on the premixer walls). To further complicate matters, fuels with larger hydrocarbons have dramatically lower autoignition times, meaning that the time available for premixing is further restricted by the potential for autoignition. Because of these complications, NO, emissions from dual-fuel LPP combustion are typically 50–100 ppmv (15% oxygen) on liquid fuel. (See Hoffman et al. [64] for a recent example.) Laboratory studies show that idealized test cases can produce significantly lower NO, emissions [65], but robust designs to produce ultra-low NO, emissions (i.e. less than 10 ppmv) are not commercially available at the present time.

An alternative approach to LPP combustion of liquid fuels is to reform the liquid fuel to a gaseous fuel prior to injection. In this scenario, the fuel injector must handle two different gaseous fuels; this is easier than handling a gas and liquid. Wang et al. [66] presented experimental data demonstrating the performance of a liquid fuel reformer suitable for stationary turbine operation. In rig testing, NO, emissions less than 20 ppmv were achieved at turbine operating conditions. The reforming process requires mixing steam with the liquid diesel fuel, and keeping the fuel line hot enough to prevent condensation of the vaporized fuel. These added complications may be offset by the advantage of producing very low NO, emissions.
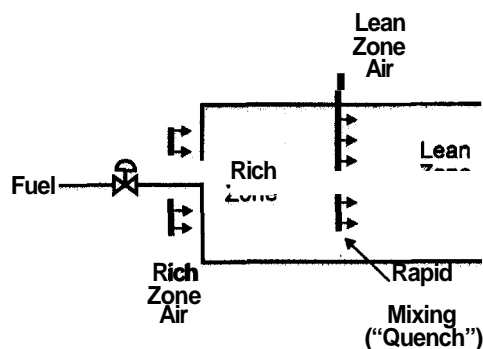
Fig. 7. Rich-quench-lean combustion system.

2.4. Fuels with nitrogen species

The preceding discussion on NO, formation has assumed that the fuel does not contain nitrogen species. Growing interest in alternate feedstocks, such as landfill gas or gasified biomass, adds a different element to the problem of fuel variability. Fuel nitrogen is typically found in the form of ammonia. When present, ammonia is typically converted directly to NO, in fuel-lean combustion. Therefore, the whole concept of premix combustion is unsuited for low NO, operation where fuel nitrogen is present. In these situations, the most common proposal for achieving low NO, is rich-quench-lean combustion. Fig. 7 shows the process. Air and fuel are burned at overall rich conditions in the first stage of the combustor. By maintaining the ammonia in a high-temperature reducing environment, much of the ammonia is reduced to N_2, Rapid mixing with air in the second-stage combustor completes oxidation of the first-stage products. Because high-temperature, flammable gases are mixed with air in this stage, the potential for diffusion flame burning is again present with associated NO, formation problems. For low heating-value fuels, this is not a major problem because the fuel is diluted. Careful design to reduce the residence time at high temperatures will minimize NO, formation. With appropriate mixing strategies, rich-quench-lean designs have demonstrated good NO, performance, at least for low heating-value fuels. Various studies [61,67,68] report tests of rich-quench-lean combustors. Conversions of fuel ammonia as low as 5% were reported. In two of the three studies, it was shown that the presence of fuel methane greatly increased the conversion of fuel ammonia to NO,. Thus, if a gas turbine relied on rich-quench-lean combustion to control NO, then the fuel stream methane content should be as low as possible. This is again a restriction on fuel flexibility.

2.5. Trace species and particulate control far gasification of coal and biomass

This paper has only dealt with rhe combustion aspects of using various fuels in low-emission applications. An equally important consideration is the effect of fuel contaminants on engine hardware. For example, coal or biomass gasification will be accompanied by vaporization of trace alkali metal species, which are very detrimental to turbine hardware. These trace species must be scrubbed or condensed from the fuel stream prior to use in an engine. Ash particulate must likewise be removed from fuel gas. For biomass applications, the presence af heavy tars in the gas stream also demands fuel treatment to avoid deposits in the gas turbine fuel system. These practical problems have been the subject of considerable research and development for both Integrated Gasification Combined Cycle (IGCC) and more recently for Biomass Integrated Gasification Gas Turbines (BIG-GT).

2.6. Summary of fuel-flexible turbine combustor requirements

In summary, a fuel-flexible combustor must avoid flashback, autoignition, and combustion dynamics, independent of fuel type or fuel impurities. Changes in fuel composition should not produce changes in static flame anchoring that could overheat the fuel injector or blow out at desired operating and upset conditions. All of these requirements must be achieved in low-emission operation. Special consideration must be given to fuel nitrogen species (ammonia), which will readily convert to NO, during lean combustion.

3. Fuel effects in reciprocating engines for stationary power

At the current time, large natural gas reciprocating engines (–1 MW) can produce thermal efficiencies of approximately 40% with NO, emissions around 1.3 g/kW-hr. As with gas turbines, high-efficiency, low-emission reciprocating engines are also affected by varying fuel properties. An example is the NG compression market, which is dominated by reciprocating engines. In the early 1990s, as advanced engines with higher operating pressures were introduced, the need for more accurate engine control over eroding knock and lean-misfire margins (discussed below) became apparent. These operating margins are affected by changes in fuel type and ambient operating conditions. Engine reliability and emission performances are compromised when combustion conditions exceed the knock or lean misfire margins. Typically, fuel-related performance problems are identified as "combustion problems" by monitoring for exhaust stoichiometry, knock or lean misfire. At the present time, fuel analysis and so-called feed-forward control strategies are seldom used to modify engine controls to accommodate fuel variability. Advancements in NG engine control techniques may provide a path for fuel-flexible engine operation with low emissions and high efficiency.

In the sections that follow, the physical aspects of engine

knock and lean misfire are reviewed, and then the various types of reciprocating engines that are typical of stationary power applications are discussed. The focus of the discussion will be the spark-ignition engines that have become a popular choice for high-efficiency gas fuel applications.

3.1. Engine knock

Of all undesirable engine phenomena, knock is potentially the most damaging. As the in-cylinder flame propagates across the combustion chamber, the end gas (the unburned mixture ahead of the flame) is compressed, causing pressure, temperature and density to increase. Some of the end-gas mixture may undergo reactions that spontaneously and rapidly release a large part of their chemical energy at rates that may exceed 20 times that of normal combustion [69]. High frequency pressure oscillations occur inside the cylinder, which produce the sharp metallic noise called knock. Hot spots within the cylinder may also lead to preignition or knock by increasing the rate of precombustion reactions in the end gas regions. The presence or absence of knock reflects the difference between the speed of the advancing flame front and the precombustion reactions in the unburned end gas. Knock does not occur if the flame front consumes the end gas before the reactions have time to reach autoignition conditions. Knock usually occurs under wide-open throttle operating conditions, and is therefore a direct constraint on engine performance. To improve knock margin, designers often compromise engine efficiency by limiting the compression ratio, where as field operational control is often done by timing retardation via feedback control using any of a variety of possible sensors. The occurrence of knock and its severity depends on the engine and the knock resistance (octane or methane number) of the fuel. Knock can be relatively harmless, or in some cases, or very damaging. When knock is very heavy, substantially more heat is transferred to the combustion chamber walls via enhanced convective heat transfer. The result is rapid overheating of the cylinder head and piston. Under these conditions, knock becomes an unstable phenomenon. The overheating further advances the ignition event, which in turn increases the intensity of knock. Knock intensity increases at a runaway rate, which can lead to engine failure in minutes. Even without a runaway process, knock damage over extended time periods leads to erosion and pitting of piston crowns, cylinder liners, rings, and valves.

For liquid fuels, the knock resistance is expressed by the familiar octane or cetane numbers. For fuel gases, the methane number measures knock resistance. The measurement range is defined by methane at one end of the scale, which has very high knock resistance, and by hydrogen at the other end, which has a very low knock resistance. Pure methane represents a methane number of 100, corresponding to a research octane number of about 140, while pure hydrogen has a methane number of zero. Another scale,

called the butane number (BN) method, is based on molar blends of methane and butane, with neat methane equal to 0 BN. The relationship between carbon number and butane number is much more non-linear than either the octane or the methane number scales '70l.

3.2. Lean misfire and cycle-to-cycle variations

The physical description of lean misfire is straightforward. On any given engine cycle, lean misfire occurs when the flame fails to light or is extinguished before releasing most of the fuel energy. The resulting effect on emissions and efficiency is obvious: UHC and CO rise to very high levels while efficiency is lost. The reason why some cycles fail to ignite or burn completely is the subject of ongoing research. Ozdor et al. [71] reviewed the literature associated with cycle-by-cycle variations. Variations between cycles may result from irregular fluid motion in the cylinder, variations in fuel/air ratio between subsequent cycles, and many other factors. Design and operating parameters, as well as fuel properties affecting lean-misfire behavior, are discussed in Section 3.4.2.

3.3. Engine type

Gas-fired reciprocating engines may be either diesel utilizing glow plug, dual-fuel engines, or spark-ignited (SI) engines. Most of the discussion will focus on lean-burn SI engines because of their increasing popularity. In this context "gas" may be any gaseous fuel. Gas engines may be two-stroke, four-stroke, turbocharged, or naturally aspirated. Gas engines may also be divided into power classes of small (20–300 kW), medium (300–1000 kW), and large (over 1000 kW). These range definitions vary according to the engine markets and geographical areas.

3.3.1. Dual-fuel gas engines

Dual-fuel gas engines can operate on both gas and liquid fuels such as diesel, or heavy oil. Combustion is initiated in the dual-fuel gas engine by the high-pressure injection of a small amount of liquid fuel into the cylinder where the liquid fuel autoignites, in turn igniting the NG mixture. This process is common in large stationary engines. Sometimes a small prechamber is used to inject a pilot of NG in lieu of liquid fuel; this is currently used for relatively low speed applications because of the relatively long autoignition time of NG compared to liquid fuel. In a dual-fuel gas engine, the air/gas mixture is admitted into the cylinder during the intake process and ignited spontaneously at the end of the compression stroke by pilot injection, with the pilot usually contributing about 5–10% of the total charge energy. Currently, there is significant research into development of "micro-pilot" engines. In these engines only 1–2% liquid fuel is used for the pilot. Research is also ongoing into using engine oil as the pilot ignition source. Using the engine's own lubrication oil as a micro-pilot energy source

Table 2
Knock-limited compression ratio and thermal efficiencies for typical gases from Zareh [72]

Fuel	Knock-limited compression ratio	Approximate brake thermal efficiency (%)
Butane	8	33
Pentane	10	36
Ethane	12	38
Methane	14	39

benefits the lean-misfire limit and reduces or eliminates the environmental problem of lube oil disposal. Micro-pilots are also discussed in Section **3.4.2** below.

3.3.2. Spark ignition

Otto cycle engines are typically referred to as SI engines or homogeneous charge engines. These may be further divided into stoichiometric and lean-burn types of engines. The stoichiometric engine, as the name implies, operates with an air/fuel mixture near stoichiometric, while the lean-burn engines utilize significantly leaner air/fuel ratios. These engines operate similarly to spark-ignited gasoline engines, where air/fuel is carbureted or injected upstream of the cylinder and ignited by a spark at the end of the compression stroke. Compared to their gasoline-fueled counterparts, SI gas engines generally have lower volumetric efficiencies (i.e. the efficiency of the piston motion to aspirate the cylinder volume) because the fuel gas displaces more air, decreasing the charge density. Furthermore, evaporative cooling in gasoline engines cools the intake mixture, increasing the charge density compared to SI gas engines. In spite of their lower volumetric efficiency, SI gas engines generally have thermal efficiencies similar to or higher than gasoline engines. Because their wider knock operating "window", SI gas engine can operate at higher compression ratios that produce higher thermal efficiency. This is especially true for gases high in methane content. The higher efficiency, however, comes with a price tag; changes in fuel composition should be made keeping in mind the changes in the knock window. For example, a shift in NG composition, such as using peakshaving gas (Section **5)** with considerable propane, will have a greater tendency to knock. For comparative purposes, Table **2** (taken from Zareh [72]) lists the knock-limited compression ratio and the associated thermal efficiency for various gaseous fuels. Note the significant difference in the knock-limited compression ratio for the various fuels. This knock operating window is a function of several operating variables but may be thought of as being bound by equivalence ratio, ignition timing, and fuel composition with all other variables held constant. In this case, increasing equivalence ratio (when burning lean) or advancing spark timing bring an engine closer to the margin of the knock operating window.

3.4. Characteristics of spark-ignited natural-gas fueled engines

3.4.1. Stoichiometric engines

In contrast to lean-burn engines, design parameters and operating parameter adjustments have only a minor effect on emissions for stoichiometric engines fitted with a downstream catalyst. Because emission control is accomplished by post-combustion cleanup, fuel composition has only a modest effect on emissions performance. Stoichiometric engines are successful at using three-way catalysts to convert hydrocarbons (HC), CO and NO_x to H_2O, CO_2 and N_2, respectively. These catalysts require high operating temperatures, typically **725–950** K **(840–1250°F),** and require a narrow fuel/air mixture to maintain high three-way catalyst conversion efficiency. Three-way catalysts are mainly used for stationary engines with less than 100 kW output. Although natural gas fuel composition has a minor effect on the catalyst cleanup in stoichiometric engines, fuel type still plays a role in engine performance due to catalyst tolerance for fuel composition changes over typical composition ranges. Efficiency, mean effective pressures (MEP), knocking behavior, and engine component wear (exhaust gas temperatures, thermal load and valve wear) are of primary importance for these engines.

Thermal efficiency is limited by a narrower knock operating window due to the near-stoichiometric mixture. In contrast with the fuel effect on emissions, the narrow knock operating window requires that careful consideration be given to fuel changes that may increase knock behavior.

3.4.2. Lean-burn engines

Engine thermal efficiency is directly related to compression ratio. However, higher compression ratios increase the propensity to knock, requiring detrimental timing modifications or tighter fuel requirements with respect to methane number. Because lean combustion decreases knocking propensity, lean-burn engines allow operation at higher compression ratios, producing greater thermal efficiency. In addition, lean operation produces lower combustion temperatures, reducing NO, emissions. The availability of excess air also increases combustion efficiency, lowering CO and total HC emissions. However, when operating at lean conditions, lean misfire must also be avoided to maintain these performance and emission advantages. Furthermore, recent testing of exhaust emissions from lean-burn, large bore natural gas engines has indicated that formaldehyde emission may be a problem at lean operating conditions. Via a thorough literature review, Mitchell et al. [73] identify both fuel protection in crevices and partial oxidation as likely mechanisms for formaldehyde formation. Equilibrium analysis does not support formaldehyde formation. Initial temperatures must be high enough to form formaldehyde, followed by rapid quenching so that destruction reactions do not take place. Mitchell et al.

suggest that fuel evolving from crevice volumes late in the piston stroke could contribute to these mechanisms.

In lean-bum engines, as concentrations of gas components with lower methane numbers increase, decreasing overall methane number, the knock limit will be approached and subsequent adjustment of operating parameters is critical. Derating by reducing the compression ratio has been the historical design method of choice to accommodate low methane number gases. This derating can lower efficiency to potentially unacceptable levels. As described in Section 3.7, advanced control methods are a promising alternative method to handle lower methane number fuels.

Because combustion in lean-bum engines is deliberately weak (i.e. diluted with excess air), lean misfire is a potential problem. Several factors affect the lean misfire limit. These factors include in-cylinder air motion, homogeneity of the cylinder charge, available ignition energy, fuel gas composition, the mixture temperature at ignition, the amounts of residual fraction and the ambient humidity. Kubesh et al. [74] investigated these phenomena in an effort to increase lean operating limits of an in-line six-cylinder, 8.1 liter displacement heavy-duty highway engine. They reported that industry efforts to address lean-misfire have focused on in-cylinder modifications to increase turbulence levels, novel piston bowl shapes that generate squish flows or enhance the effects of swirl, and high-energy ignition systems to extend the lean flammability limit (but not without significantly decreasing spark plug life). In their applications, Kubesh et al. assumed that fuel variability will be minimized in transportation applications. Thus, stationary applications were not considered in their analysis. Mixture preparation temperature was determined to have a moderate effect on the lean flammability limit, see also Quader [75]. For an engine operating with propane–air mixtures, the lean limit equivalence ratio decreased by approximately 0.0177 for each 56 K increase in compression temperature. Increases in specific humidity were detrimental to the engine lean limit. In their testing, the lean misfire limit equivalence ratio increased by 0.030 when the specific humidity increased from 0.008 to 0.019 (kg_{water}/kg_{air}).

Ozdor [71] reviewed the literature regarding cyclic variability in spark ignition engines. Near the lean misfire limit, cyclic variability increases, making this literature review relevant to the problem of lean-misfire. Their findings point to the importance of a strong ignition source in extending the lean misfire limit of spark ignited engines. Pilot ignition sources are used extensively in engines utilizing many fuel types. The pilot aids combustion initiation with stubborn fuels and extends the lean misfire limit with natural gas or low-Btu gas. Pilot fuel combustion can use the same fuel, or a different fuel, than the primary combustion. In either case, the pilot tends to contribute to poor emission performance above a threshold quantity of pilot fuel. The term "micro-pilot" is often used to describe an engine which uses minimal pilot fuel. With small pilot fuel quantities, the benefits of extending the lean flammability limit outweigh

the detrimental effect on emissions. In one recent study of a natural gas engine [76], a spark igniter was replaced with a diesel fuel micro-pilot providing less than 1% of the total fuel energy. This arrangement produced a 50% reduction in NO, compared to the spark-ignition configuration. Other types of pilot and main fuel combinations (e.g. natural gas piloting in a lean-burn NG) may allow higher pilot levels while still balancing lean-misfire and emissions benefits.

The thermal efficiencies of NG-fueled lean-bum engines are typically a factor of 1.2 times those of stoichiometric engines. However, NO, emissions are typically an order of magnitude greater because downstream lean NO, catalysts are not available (Section 3.5). Increasing the lean-misfire limit can lower NO, emissions by increasing air/fuel ratio and decreasing combustion temperature. Using syngas or hydrogen-enriched gas can extend the lean-misfire limit because hydrogen has a lower energy requirement to initiate combustion than natural gas. Hydrogen also has a higher laminar flame speed and can burn much leaner than any other gas. Smith and Bartley [77] studied the effects of NG/synthesis gas mixtures on exhaust emissions and performance of a modified single-cylinder Caterpillar 1Y540 research engine. A rhodium catalyst on an alumina substrate was used to conduct partial oxidation reforming of the natural gas fuel, producing a syngas that contained up to 30% hydrogen. The authors found that NO, was reduced to below 20 ppm at greater than 32% efficiency without significant cyclic variability.

3.5. Emission control in lean-bum engines

3.5.1. Selective catalytic reduction and advanced catalytic converters

In contrast to stoichiometric engines, a three-way catalyst will not remove NO, at lean fuel/air ratios (but it will oxidize hydrocarbons and CO). Thus, as in gas turbines (which always operate with excess air), SCR is sometimes used for NO, control in large lean-bum engines. The operating issues associated with SCR on reciprocating engines are the same as those discussed in Section 2.1 for gas turbines. SCR catalysts are expensive and bulky; therefore, they are only potentially desirable for larger installations of over 1000 kW.

At the present time, there is considerable research aimed at developing oxidation catalysts for lean-bum engines. These catalysts are designed to react hydrocarbons present in the exhaust selectively with NO, rather than with oxygen. Typical of all such catalysts, NO, conversion to N_2 is poor at low temperatures, rises to a maximum at an intermediate temperature, and then falls as the temperature increases further. Maximum NO, conversion generally takes place close to the temperature at which the hydrocarbons light off. The temperature at which the maximum NO, conversion occurs is a function of the active metals used in the catalyst. Platinum-containing catalysts peak in the range of 475–525K (390–480°F), while the maximum for

Table 3
Approximate fuel gas properties from Zarah [72] and Lefebvre [7]

Gas type	Main constituents	Density (kg/m^3)	Flammability limits in air volume (%)		Heating value (MJ/m^3)		Typical a/f ratio (volume)	Methane number	Wobbe index (MJ/m^3)
			LOW	High	LOW	High			
Methane	CH_4	0.72	5	15	36	38	9.7	100	48
Ethane	C_2H_4	0.78	3	13	65	67	17	44	63
Propane	C_3H_8	0.52	2.2	11	89	94	24	32	75
Butane	C_4H_{10}	0.39	1.8	8.4	116	126	31	10	80
Carbon monoxide	CO	1.87	12.5	75	13	13	2.4	62	13
Hydrogen	H_2	0.09	4	75	11	13	2.4	0	41
NG	$CH_4/C_2H_6/CO_2/C_3H_8/N_2/O_2/H_2$	0.74	4.8	14	36	42	10	85	40
Sewage gas	$CH_4/CO_2/N_2/O_2$	1.05	Relatively narrow		21	26	5.8	140	22
Landfill gas	$CH_4/CO_2/N_2/O_2$	1.1	Relatively narrow		16	20	6.6	160	15
Gasifier gas	$N_2/CO/H_2$	1.1	≈5	≈32	5	7	1.1	Low[a]	5

[a] High CO and H_2 content

copper-containing catalysts is in the range of 725–775 K (840–930°F). The major problems faced by lean NO, catalysts are the narrowness of the temperature operating window, the limited NO, conversion even at optimal temperature, and the need for specific hydrocarbons for the selective reduction of NO,. The inherently low levels of hydrocarbons in the exhaust from lean-burn engines can require that the exhaust hydrocarbons be enhanced by the addition of fuel during either the expansion or exhaust strokes. Impurities of sulfurous compounds in NG in the form of H_2S can directly form sulfate compounds in the presence of excess oxygen. The platinum catalysts have the additional shortcoming of being highly active for sulfate formation. Sulfates can remain on the catalyst surface masking the catalyst from further reactions. Phosphorous, sulfur and zinc compounds found in engine lubricants can also contribute to catalyst fouling.

3.5.2. Exhaust gas recirculation

It has been shown that exhaust gas recirculation (EGR) is a very effective method for NO, reduction in lean-burn engines. The effectiveness stems from two mechanisms. The first is simply that the recirculated gases, which are higher in CO_2, act as a heat sink. Heat absorbed by the gases is directly proportional to the product of the EGR flow rate. The second and more important mechanism is that of displacement of some of the oxygen in the fresh *air* charge. In a simplified way, the formation rate of thermal NO, is a function of the concentrations of N_2, O_2, residence time and combustion temperature. Considering the dissociation of NO and NO_2, this may be expressed as

$$\frac{d[NO]}{dt} = K_1[N_2, O_2] - K_2[NO, NO_2]$$

where K_1 and K_2 are reaction rate constants that are strong functions of combustion temperature. Controlling the temperature or residence time is important because the temperature is a function of the initial equivalence ratio and the EGR rate. The concentration of oxygen in the combustion mixture is affected by the EGR.

In general, EGR affects the engine by:

- reducing the flame temperature and, thus, NO, by increasing the heat capacity of the cylinder charge;
- affecting ignition delay through two competing mechanisms: the increased intake temperature reducing ignition delay, and the reduced O_2 concentration from dilution with CO_2 and N_2 increasing ignition delay;
- affecting the duration of combustion by influencing the intake oxygen concentration, the stoichiometric flame temperature, and flame structure.

As part of an engine control strategy, EGR can aid in knock control as well as control at the lean-misfire limit. In situations where fuel variability is a problem, EGR control represents another control variable. For lean-burn engines, the effect of EGR on engine performance and efficiency is similar to the addition of excess *air,* but with the added benefit of reducing the oxygen available for NO formation.

3.6. Changing methane numbers

The most commonly used gas fuels are NG, liquefied petroleum gas (LPG) and sewage or landfill gases. The use of sewage and landfill gases for reciprocating engines is gaining interest. Methane and inert gases (CO_2 and N_2) are the main constituents of sewage and landfill gas. LPG gases are either propane or butane, or mixtures of the two. These are easily stored and transported due to their low liquefaction temperature (relative to methane). Table 3 shows typical and significant properties of gaseous fuel **types,** including the methane number. Included in Table 3 are typical sewage, landfill and gasifier gases. Sewage and landfill gases show a high methane number and, thus, a large knock operating window. Gasifier gas is not often used as an engine fuel, but with the advent of clean coal gasification systems and petroleum coke-fired gasifiers for chemical feedstock generation, it is becoming more prevalent. It should be noted that because of its high H_2 and *CO* content, gasifier gas will provide a rather narrow knock operating window but extended lean misfire limit.

For peakshaving (see Section 5), gas companies or utilities may mix substantial quantities of propane/butane to natural gas pipelines. Blending natural gas with propane/butane substantially reduces knock resistance. Schiffgens et al. [78] report that natural gas methane numbers in Germany range from 70 to 90 with numbers as low as 25 reported for peakshaving conditions.

Typically, design engineers are asked to focus on designing a gas engine such that optimized efficiency, power and emission behavior can be obtained for a specific range of base gases, Large engine manufacturers request base fuel composition as part of the information for their bid package. This enters heavily into the warranty and power guarantee. It must be ensured that even with changing gas properties the engine will operate safely, preventing any damage to engine components due to increased knock. In addition, acceptable efficiency and power must be obtained while complying with emission limits. Designing the engine for a conservative methane number gas may not always be a practical solution. As noted in the next section, advanced engine controls offer a method to compensate for variable fuel properties,

3.7. Engine controls

When using gaseous fuels where engine knock limit is not reached, it would be helpful to have a methane-number fuel sensor for control of lean-burn engines to ensure high loads and efficiencies while maintaining acceptable emissions. When the knock limit is reached, due to decreasing methane
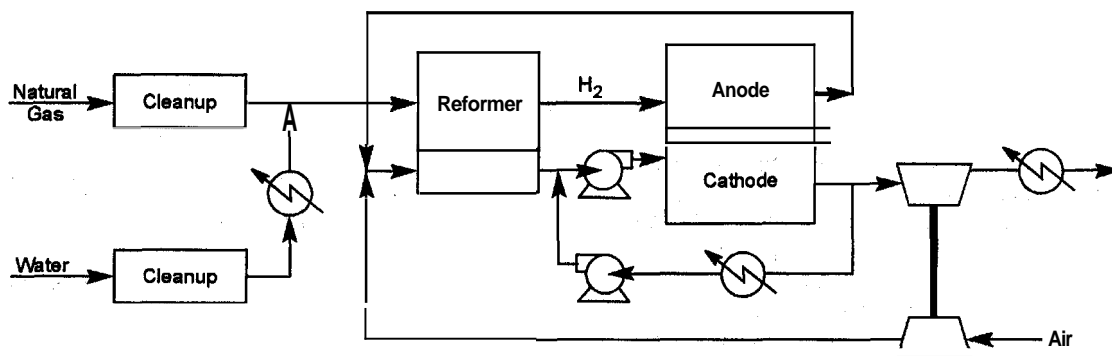
Fig. 8. Schematic of fuel-cell.

number, engine control could switch to knock control. To obtain high efficiency and high cylinder pressures, knock-controlled and lean-misfire-controlled operation is necessary over essentially the entire range of methane numbers. The current approach is to simply monitor for knock or misfire and adjust the timing or engine stoichiometry when they occur.

3.7.1. *Knock detection and control*

Many methods of knock detection have been used. Leiker et al. [79] designed a novel knock meter in 1972 that is still commonly used. It monitors cylinder pressure and filters out the pressure rise due to compression; it then examines the remaining signal for pressure oscillations. A high intensity flash, along with increased ionization, occurs with knock. As a result, optical probes and ionization probes have also been used as knock detectors.

With the increased ratings and power levels demanded from NG lean-burn engines, hock detection is an area of increasing interest. As engine control capabilities have become more sophisticated with closed-loop, lean-burn equivalence ratio control, engine knock occurrences in the field have become much less frequent on the more advanced lean-burn engines. But with heavy-duty gas engine ratings and efficiency targets increasing, engines are calibrated to operate with lower knock operating margins. Therefore, the ability to detect engine knock conditions is of great interest. Podnar et al. [80] investigated various knock sensors on a heavy-duty NG-fueled 8.1 liter engine to determine their optimal configuration in terms of the number and location of sensors and required signal conditioning. Among the knock sensors tested were various commercially available piezoelectric knock sensors, as well as a magnetostrictive sensor. These authors concluded that the optimal sensor number and placement is highly engine-specific.

In general, knock control capability would allow more aggressive ratings to be developed for heavy-duty NG engines and would result in increased durability and decreased performance uncertainty.

3.7.2. *Misfire detection and control*

In a lean-burn NG engine, misfire is of increased interest since these engines are calibrated for operation near the lean misfire limit. Podnar et al. [80] included misfire detection and control as part of their investigations. Two types of misfire detection technologies were evaluated during the project. Both in-cylinder and exhaust ionization probes were demonstrated as capable of misfire detection, but the exhaust probe was easier to install, and less expensive.

Misfire detection and control, combined with knock control and equivalence ratio control, are the techniques used in modern NG engines for transportation applications. Combinations of these techniques are becoming common for stationary engines as well. NG fueling for transportation requires a constant fuel specification in light of the control requirements over highly varying speed and load conditions. In stationary applications, however, fuel is often highly variable. As noted above, improvement to the situation warrants investigation of online gas composition measurements for feed-forward control in stationary engines.

3.7.3. *General engine control strategy*

Schiffgens et al. [78] investigated changing methane number effects on both stoichiometric and lean burn spark ignited natural gas engines. They developed a new gas engine control device and strategy based on their findings. The input variables to their control system include methane number sensor output, engine load, crankshaft position and input from a knock sensor. Outputs include signals to control ignition timing and air/fuel ratio. The engine is precontrolled by a methane number sensor based on fuel gas thermal conductivity to maintain acceptable emission levels with fuels of varying methane number ranges before the knock limit is reached. In their control scheme, the air/fuel ratio is leaned as a first response to decreasing methane number as indicated by the methane number device. Secondly, with further decreasing methane number, ignition timing is adjusted (retarded) based on input from the methane number fuel sensor. These adjustments must

maintain knock margin while still maintaining appropriate NO levels. When the methane number falls sufficiently further, the methane number control is switched to knock sensor control to ensure that engine damage does not occur. Timing is further retarded or load is reduced until knock free operation is established.

3.8. Summary of issues for fuel-flexible reciprocating engines

Operation of high-efficiency reciprocating engines is a careful balance between efficiency, engine durability and emissions. To achieve high efficiency, both stoichiometric and lean burn engines operate near the knock margin and lean burn engines near the lean-misfire limit. Both the knock and lean-misfire margins depend on the fuel type, placing a restriction on performance if a given engine is to meet performance expectations. Although very lean operation can reduce NO levels, further progress will likely require some method to enhance lean combustion, such as micro-piloting, or using hydrogen generated by reforming a portion of the natural gas. In lieu of significant advances in the combustion process, exhaust stream cleanup may be required to meet aggressive NO goals. Further improvements in engine performance may benefit from engine controls that actively monitor fuel composition and take needed control actions to allow operation at lean conditions without misfire.

4. Fuel cell power systems

It is anticipated that fuel cell systems will play an increasing role in power generation, particularly as the distributed power market begins to expand. Currently, there are four basic types of fuel cells under development: phosphoric acid fuel cells (PAFC), polymer electrolyte fuel cells (PEFC), molten carbonate fuel cells (MCFC) and solid oxide fuel cells (SOFC). Presently, PAFC have the most field experience through ONSI, a subsidiary of United Technologies Corporation. Their 200 kW (PC25™) systems entered commercial operation in 1992, and since that time have acquired over 2 million operating hours. The other fuel cell types have developers planning to bring commercial systems to the market within the next 2–3 years. The widely recognized benefits attributed to fuel cells are their superior emissions performance (<1.0 ppm NO), and high efficiency ($>50\%$). Another benefit not commonly discussed is their ability to be *reliably* fuel flexible. This section discusses the general operation of fuel cells when using the typical hydrocarbon-based fuels currently available on the market; e.g. natural gas, JP-4, naphtha and methanol, etc., as well as other opportunity fuels, such as landfill gas or digester gas.

4.1. General fuel cell system operation

The major components of a fuel cell, regardless of type, are fuel cleanup, fuel reformer and fuel cell stack (anode and cathode). Fig. 8 shows a generic representation of a fuel cell system, with *independent* fuel reforming, using heat from a burner to reform the fuel. As explained below, fuel reforming can also be integrated into the fuel cell stack. The fuel cleanup unit removes any fuel-gas sulfur compounds which can poison the reformer and fuel cell anode electrodes; e.g. forming nickel sulfide in certain SOFC systems. The platinum catalyzed electrodes in PEFC systems are also poisoned by carbon monoxide produced from the reforming process. Presently, most PEFC systems require that CO be removed to levels below 10 ppm, although some developers are attempting to produce improved CO tolerant electrodes which can handle CO concentrations higher than 50 ppm. For MCFC and SOFC systems, as well as PAFC systems when operated correctly, carbon monoxide is acceptable. This results from the fact that these systems operate at high enough temperatures where CO can be oxidized reformed if present on the electrode.

For some fuel cell types, the reformer can be made integral, or partially integral with the fuel cell to achieve improved overall system efficiency. While both the fuel cleanup and reformer units are fairly simple and reliable, both systems must be operated to avoid conditions where carbon deposition may result. Such conditions lead to deactivation of the cleanup and reformer catalysts, as well as the down stream anode electrode. In some fuel cell configurations, the fuel cell exhaust contains residual fuel gases which must be oxidized. In these cases, an exhaust cleanup unit is installed downstream of the fuel cell, often using a catalyst-based oxidizer to react the remaining fuel. Finally, for most applications, a power conditioner takes the DC power generated by the fuel cell and provides controlled AC power output.

4.2. Fuel cell governing parameters

For purposes of the present discussion, the parameters that determine the performance of the fuel cell unit are the input concentrations of reactive components (e.g. H_2 and CO at the anode, O_2 at the cathode), energy dissipation, and overall reactant utilization. While fuel variability can affect the latter two parameters when trying to maintain fixed output power, the main impact to the fuel cell performance will be due to a change in voltage resulting from the change in reactant concentrations. This effect can be described via the so-called Nernst equation, which shows an improvement in fuel cell output voltage as reactive concentrations increase. As an example, for SOFC systems using pure H_2 as a fuel, the Nernst equation is written as:

$$E = E^0 + \frac{RT}{2F} \ln \frac{[H_2][O_2]^{0.5}}{[H_2O]}$$

where E is the reversible voltage, E^0 is the voltage at standard state conditions, R is the universal gas constant and F is Faraday's constant. The Nernst equation predicts the ideal system performance, and can be assessed for any system using an equilibrium analysis that allows the electrochemical conversion of the reactant species. Similar Nernst equations can be derived for other fuel cells, and for other $H_2 + CO$ fuels [81]. The equation makes it clear that any fuel variation resulting in H_2 and CO concentration fluctuations will cause a change in output voltage, It can be shown that the comparison between the actual cell voltage and the ideal voltage (determined from a Nernst equation) directly relates to the efficiency of the electrochemical conversion process, and therefore, directly relates to the system fuel efficiency.

4.3. Designing an optimal, stable and flexible power system

When a power system is designed for peak efficiency at the lowest lifetime cost, tradeoffs are usually encountered that force the system to operate in a certain control parameter space. In addition, only one fuel can be used to achieve this peak efficiency, Employing other secondary fuels normally results in greatly reduced efficiency, reliability and operating range. For fuel cell systems, however, the development of a reliable, fuel-flexible system can be accomplished with relatively little effort during the design stage. The designer will simply be required to account for any new thermal demands/dissipations due to the variation in fuel, and ensure that the reformer and fuel cell continue to operate effectively. Clearly, each fuel type will result in its own performance curve, and some fuels will be able to provide higher current and/or efficiency than others, but the risk of entering an unstable/dangerous mode of operation is very small. **This** stability feature of fuel cells is largely attributed to the fact that the distribution of flow over the many cells prevents the entire mass of flow to experience coherent (high energy) instability, and the fact that the major fuel cell processes (reactant gas delivery and electrochemical conversion) are sufficiently decoupled. That is, the electrochemical rates may depend on gas delivery rates, but gas delivery rates do not depend much on the electrochemical rates. This is not the case for other power technologies, such as IC engines or gas turbines, in which chemical reactions and flow dynamics/turbulence are integrally linked, or where distinct flow features have similar time scales and can become mutually excited/enforced (e.g. vortex mixing and acoustics).

As an example, to accommodate changes **from** a pure methane fuel to a fuel with methane plus significant amounts of H_2 and CO, very little needs to be considered outside of the well-understood impact on system operation and performance; i.e. the addition of H_2 will decrease the energy density of the fuel supply, forcing a change in control valve position in order to maintain constant power output. Other than these types of changes and the resulting impact to

system voltage (efficiency), the operation of the fuel cell will be mostly unchanged. In oontrast, if a gas turbine or reciprocating engine incurred the same fuel change, it would likely require significant modifications to avoid flashback, or unstable operation.

4.4. Alternative fuels development and market status

Most current applications of fuel cells utilize natural gas as a fuel source. However, propane gas, digester gas from sewage treatment plants, landfill gas and waste hydrogen can also be used. In fact, the phosphoric acid systems cited above are already advertised to operate with all of the above-mentioned fuel types.

To support the penetration of fuel cells to these markets, a growing number of papers have been published that describe the development of different fuel cells for alternative fuels. Aside from the additional cleanup equipment due to the higher moisture, sulfur and halogenated compounds, the modifications may include a larger fuel control valve and fuel control venturi, plus resizing to fixed orifices. Geyer [82] describes the operation of a fuel cell developed by Toshiba for breweries. About 5% of a brewery plant's power can be provided by such a fuel cell. Similar examples of alternative fuel use can be found in Refs. [83–87].

In spite of these "fuel-flexibility-benefits" offered by fuel cells, fuel cells have not been able to penetrate much of the power generation market due to their relatively high costs compared to turbines and reciprocating engines. Therefore, the initial market for fuel cells is likely to be limited to areas with tight emission regulations, or where grid electric power is more expensive than what can be produced on site. Current costs for the PC25 systems are about \$3000 per kW. Projections for fuel cell costs in the next 3–4 years are around \$1200 –1800 per kW versus ~\$600 per kW for micro gas turbines (i.e. systems between 30 and 250 kW), and reciprocating engines. The cost of fuel cell systems, however, continue to have the greatest potential for being further reduced, especially given the fact that fuel cells continue to command much attention by both research organizations and commercial developers.

5. Natural gas properties

The intent of **this** paper is to identify the technical issues associated with developing low-emission energy systems capable of using different fuels. The potential sources and properties of gaseous fuels are widely variable (see Tables I and **3**) so that detailed discussion **of** any one fuel is likely to be applicable to just a site-specific application. For **this** reason, **this** paper focuses on the technical issues associated with using different fuels, rather than a detailed discussion of using a specific fuel. However, because natural gas is the most common gaseous fuel, in **this** section we review the

Table 4
Natural gas composition based on methane fraction from Schaedel et al. [88]

Constituent	10th Percentile mol%	Mean mol%	90th Percentile mol%
Methane	83.94	93.05	95.98
Ethane	5.62	3.47	2.14
Propane	0.99	0.67	0.36
C_4 and higher	0.31	0.33	0.38
Nitrogen	6.28	1.67	0.53
Carbon dioxide	1.37	0.81	0.73
Oxygen	1.49	−0	−0
Wobbe number	1225	1332	1347
HHV (Btu/scf)	982	1029	1029
Specific gravity	0.64	0.6	0.58

properties of natural gas. The discussion covers the domestic natural gas market in the United States.

The NG pipeline network in the United States is highly integrated, with gas flowing from production sites through high-pressure pipelines to the local distribution companies. According to Schaedel et al. [88], the composition of gas arriving to the customer is affected by producing field characteristics, proximity to gas supply centers, number of different pipelines serving the distributor and how and where each gas source has been processed.

In fact, gas transported to a distributor has a unique chemical and physical "signature" that may differ from gas delivered to other utilities in other geographical regions. Before deregulation of the gas industry, purchase contracts between pipelines and producers mandated the quality of gas owned and transported by pipelines [89]. Since deregulation, gas transmission companies must carry gas from various producers, and the compositions can vary between their gas and the "non-owned" gases they transport. Typical contract limits are specified for selected NG components such as corrosives, inerts, oxygen, hydrogen, water and heating value, but not hydrocarbon content. According to Attari and Chao [89], "nearly all pipeline gas is consumed in combustion equipment that allows consid-

erable variation in gas composition provided that its overall energy content or heating value remains within a relatively narrow band."

The Gas Research Institute (GRI) has recently studied gas composition in major market centers over a broad range of geographic locations to quantify the potential regional and seasonal variations in NG composition, and the effect these variations might have on stationary engines and NG vehicles [90]. Over 6800 gas analyses were obtained from 26 cities located in 19 states and each of the major geographical regions of the United States. Major urban areas were chosen from different regions of the country based on their size, potential for NG vehicle use, and the city's status as a non-attainment region for ambient emissions (primarily ozone). Statistical analysis of the data indicate that NG is primarily methane, ethane, propane and inert gases, with relatively low levels of butane or heavier hydrocarbons. The analysis included calculating a number of physical properties for each sample point (heating value, specific gravity, Wobbe index, methane number and critical compression ratio); determining the mean, minimum, maximum, loth and 90th percentile for each city's data; then calculating statistics on a weighted national basis (based on the volume of NG delivered in each individual city). The weighted national statistics are summarized in Table 4, taken from Schaedel et al. [88].

The results indicate that gas characteristics are reasonably consistent, but subject to significant localized exceptions. Localized exceptions can result from the following:

- local gas production where heavier hydrocarbons such as propane and ethane are produced along with NG from oil-producing reservoirs;
- local gas sources such as refinery gases or landfill gases;
- elevated levels of higher hydrocarbons because of insufficient blending;
- insufficient removal of higher hydrocarbons from processing plant upsets;
- "peakshaving," blending of non-methane gases (primarily propane) with air to meet peak winter demand.

In addition to fuel composition variation, elevated

Table 5
Representative peakshaving mixtures: air/propane/natural gas from Schaedel et al. [88]

Constituent (mol%)	Base gas	10% Peakshaving	15% Peakshaving	20% Peakshaving	25% Peakshaving
Methane	94.69	85.22	80.48	75.77	71.04
Ethane	2.93	2.68	2.55	2.42	2.30
Propane	0.22	3.85	5.44	7.48	9.29
C_4 and higher	0.05	0.13	0.19	0.21	0.25
Nitrogen	1.44	6.04	8.53	10.63	12.93
Oxygen	0.0	1.26	1.94	2.52	3.15
Carbon dioxide	0.67	0.60	0.57	0.54	0.50
HHV (Btu/scf)	1015.6	1014	1007	1013	1012
Wobbe number	1329.5	1262	1225	1203	1177

levels of water vapor can be encountered in NG. Normal pipeline levels of water vapor are less than 7 lb/million scf. This can be exceeded due to several factors, including equipment failures in gas dryers at flooded underground storage facilities, occasional failures in upstream processing equipment and leaks in low-pressure piping that passes below bodies of water or the local water table.

Local distribution companies rely on various sources of supplemental gas to meet short-term peak gas loads. These sources include using gas from underground storage, revaporizing stored liquid NG and blending propane and air with pipeline gas. Propane/air peakshaving blends are adjusted to approximate the heating value and Wobbe index of the NG in the distribution line while the oxygen in the dilution is kept below the flammability limit. Blending is necessary since propane has about twice the volumetric heating value of NG (see Table 3). Table 5 shows the mixture composition by mole percents of constituent gases for the given base NG at varying levels of peakshaving. Notice that the increased concentration of propane and nitrogen becomes significant with increasing levels of peakshaving. Methane levels also fall significantly with increasing levels of peakshaving. A survey conducted by the American Gas Association [91] and later updated by the Institute of Gas Technology [88] assessed the peakshaving practices of over 110 gas distribution companies. The following information was obtained:

- Almost half of survey respondents had abandoned propane/air peakshaving due to the increased availability of NG. The Institute of Gas Technology estimated an additional 50% reduction in current peakshaving levels by 1998.
- For the largest 15 propane/air peakshaving utilities, the average duration of peakshaving during the past 5 years was 3 days.
- The most common propane/air mixtures are **55%** propane/45% air and 50% propane/50% air.

The practical conclusion from these observations is that peakshaving is relatively rare. However, as noted throughout this paper, even brief excursions in fuel composition can create serious problems for high-performance gas turbines and reciprocating engines.

6. Conclusions

This paper has reviewed the technical issues associated with using variable-composition gaseous fuels in low-emission energy systems. We considered advanced gas turbines, reciprocating engines and fuel cells. In gas turbines, premix combustion has become a popular choice for reducing NO, emissions. Although premix combustion has a proven advantage for emissions, flame position and stability may be affected by changes in fuel composition. Flame flashback, autoignition, dynamic oscillations and lean blowout must be considered as potential complications associated with changes in fuel composition. Specific considerations are noted for medium and low heating value fuels compared to natural gas and fuels containing ammonia, which are typical of gasification.

In high-performance reciprocating engines, it is difficult to produce low emissions and high efficiency by using lean combustion. Lean operation can allow high compression ratios with lower knock potential, but the NO, emissions are still very high compared to stoichiometric engines, which can use an exhaust stream catalyst. Further reduction in lean-misfire limit has shown promise as a method to reduce NO, emissions, but it should be expected that careful operation near the lean-limit will be complicated by changes in fuel properties. For stoichiometric engines, downstream catalysts are available that significantly reduce NO, and HC emissions. However, compression ratios and engine efficiency are limited by the fuel chemistry. Because the higher hydrocarbons have a shorter autoignition time, non-methane components can initiate premature ignition, leading to damaging engine knock. The potential to monitor fuel composition and adjust engine operation to accommodate fuel variability was discussed, but no practical implementation of this concept has been developed to date.

Compared to turbines and spark-ignition engines, fuel cells can be readily designed to handle a range of gaseous fuels. In a fuel cell, the gases are reformed to hydrogen feedstock (and CO in some cases) needed for stack operation. Although the cell voltage may be modestly affected by a change in the hydrogen concentration, there are few other complications that cannot be accommodated in the system design. While the fuel reforming and cleanup adds flexibility to the fuel cell system, it also adds to the overall initial cost. For similar power ratings, the initial cost of fuel cells is currently more than four times that of comparable reciprocating or turbine options. However, in situations where fuel costs are high, and emissions regulations are very tight, fuel cells may be an attractive option to use fuels with variable properties.

The potential combination of gaseous fuel sources makes it difficult to discuss specific issues associated with any one source. However, because natural gas is the common gaseous fuel, the properties of the US domestic natural gas supply were reviewed. In most circumstances, natural gas is primarily methane, with a variable ethane content less than 20%. In the United States, the natural gas higher heating value is held close to 1000 Btu/scf. During peak gas demand, peakshaving gas may be used to supplement the natural gas supply, typically replacing 10% or more of the natural gas with a propane-air mixture. Although peakshaving is relatively infrequent, potential fuel-switching issues associated with turbines, reciprocating engines, and fuel cells should be recognized in locations where peakshaving is expected. In addition to peakshaving, gas

processing-plant upsets may introduce momentary changes in fuel composition. These changes are insignificant for conventional applications where the fuel is burned in boilers or heaters. However, for advanced energy systems, careful consideration of the effect of even brief fuel-composition excursions should be made.

Acknowledgements

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